THE HEATING AND VENTILATING MAGAZINE

1123 BROADWAY NEW YORK NOVEMBER, 1907

The Performance of Hot Blast Coils

II.

A Discussion of Mr. Burt S. Harrison's Formula and Charts, Published in the October, 1907, Issue of "The Heating and Ventilating Magazine"

EDITOR HEATING AND VENTILATING MAGAZINE:

I am greatly interested in Mr. Burt F. Harrison's charts and the formula on which the charts are based, but the author does not give the basis from which he derives the formula. I would have a great deal more confidence in the formula if I had the opportunity to verify the computations upon which it is based. I would especially like to know from what data it is found that the rise and temperature of air passing through a heater varies as the reciprocal of the cube root of the velocity, and from what data it was found that the rise varied also as the reciprocal of $8/n + .24$; also where the value of 50 was obtained and how the formula of Chart 2 is derived. The figure 1.9 in that formula, I think, should be 2.03. I figure as follows: 1-inch pipe 1 foot long is 1.315 inch, outside diameter, and has 0.344 square feet external surface. The distance between the pipes is 35/65 of 1.315 inches = .707 inch which, added to 1.315, makes 2.032 inches, the width of pipe + one space. Each 1 foot width of heater will contain 12/2.032 pipes and this multiplied by .344 gives 2.03 square feet heating surface per square foot area of section for one row of pipe.

Mr. Harrison's Chart No. 2 does not seem to agree with the table of the International Correspondence School instruction paper 760 B. I take the figures below from this table and compare them with figures taken from Mr. Harrison's chart.

The figures in parentheses are taken from Mr. Harrison's chart. It will be seen that the figures from the chart are lower than those from the International Correspondence School tables for the lower velocities and higher for the higher velocities. The authority for the Correspondence School chart is not given.

I will be glad if the author will make a statement to reconcile discrepancies between this chart and the I. C. S. table and also show that his chart is worthy of more confidence than the table.

WILLIAM KENT.

Syracuse, N. Y. Oct. 25, 1907.

ANSWER BY BURT S. HARRISON.

In reply to Prof. Kent's letter with reference to the charts and formulæ on "Performance of Hot Blast Coils," published in your October number, I will try to explain how the results were arrived at.

The data were taken from a number of tests made at different times to determine the effect of each varying condition. The most common size of coils being 5 feet x 6 feet—4-

Row	600 Vel.		1200		1800		2400	
1	9.9	(7.5)	13.1	(11.9)	15.5	(15.5)	17.6	(18.9)
12	6.6	(5.7)	8.4	(9.0)	9.9	(11.8)	11.1	(14.3)
24	5.3	(4.5)	6.7	(7.2)	7.7	(9.4)	8.6	(11.4)

row and 2-row coils of the type shown on sketch.

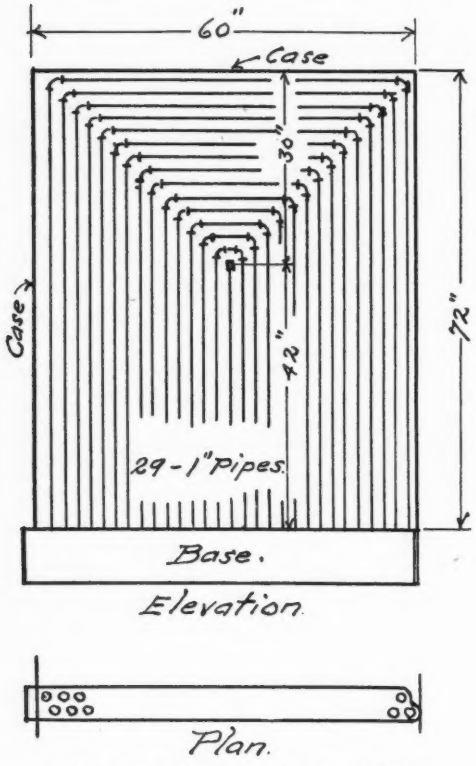
I figure the free area as follows:

Coil 60 inches long by 72 inches high inside case.

29 pipes in each row including one dead-end center pipe.

Outside diameter of 1-inch pipe, 1.315 inch.

28 spaces (C to C of pipes) = 28x 2 = 56 inches.



STANDARD 5 FT. x 6 FT. TWO ROW BLAST COIL

Center of right handpipe to case, 1.5 inch.

Center of left-hand pipe to case, 2.5 inches.

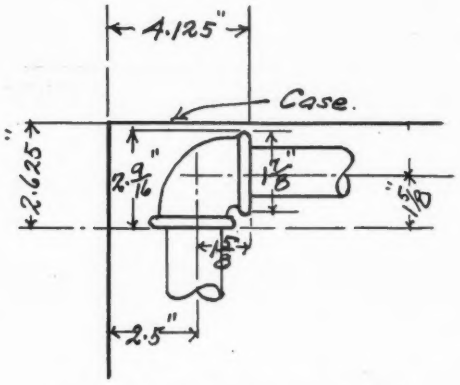
(Greater space to allow of staggered rows). Total, 60 inches.

Total free area = 60 inches—(29x 1.315) = 21.865 inches.

21.865x72 ÷ 144 = 10.9325 square feet area.

Deduct for 28 elbows having each about 1 square inch excess of area over what a similar length of pipe would have, and one cap having about .8 square inch excess of area and the free area of the coil becomes 10.9325 square feet—(28.8 ÷ 144 square feet) or 10.7325 square feet.

The superficial area of coil is 30 square feet and free area is 10.7325 ÷ 30 = 35.775% or *approximately* 35% as stated before.

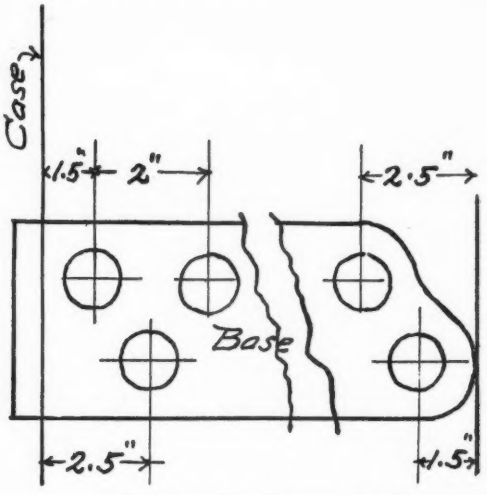


ELBOW OF PIPE IN BLAST COIL

The heating surface of the coils was figured as follows:

Vertical Pipes.

Outside pipe, 72 inches—2.625 inches = 69.375 inches.



DETAIL OF BLAST COIL BASE

Pipe next center, 69.375—26 inches = 43.375 inches.

Total, 112.750 inches ÷ 2 = 56.375 inches average length.

Horizontal Pipes.

Tap pipe, 60 inches—(4.125 inches + 3.125 inches) = 52.75 inches.

Shortest pipe, 52.75—52 = .75 inch.

Total, 53.50 ÷ 2 = 26.75 inches average length.

Center pipe = 42.00 inches.

Total pipe surface = (28x56.375) + (14 x26.75) + 42 = 1996.20 lin. inches = 166.35 lin. feet of pipe.

166.35 ÷ 2.904 = 57.28 square feet heating surface per row of pipe, and with superficial area of 30 square feet = 1.909 square feet H. S. per square foot superficial area of coil, or in round numbers 1.9 square feet H. S. per square foot of coil area.

The elbow and base surfaces are purposely omitted in order that only prime heating surface may be considered, as it is customary to specify blast coils by actual lineal feet of one-inch pipe heating surface.

The elbows and bases bear almost a constant relation to the prime heating surface of the coils. (About 10% of the actual coil surface.)

In figuring the capacity of a coil the result is the same whether only prime heating surface is figured at its full value and the auxiliary heating surface is added or the auxiliary surface figured into the calculations and the value of the prime surface reduced proportionately.

The formula for the rise in temperature of air passing through a coil is as follows:

$$\underset{\text{velocity.}}{\text{Factor of}} \quad \frac{1}{\sqrt[3]{V}} \quad \times \quad \underset{\text{temperature.}}{\text{Factor for difference in}} \quad (T-t) \quad \times \quad \underset{\text{heater.}}{\text{Factor for depth of}} \quad \frac{1}{8/N + .24}$$

No explanation is needed for the factor (T—t). The first factor was derived by varying only the velocity of the air through the heater, the other factors remaining constant.

The averages of a number of readings were found to be proportionately as follows:

Velocity in feet per minute.	Average comparative temp. rise.	Reciprocal of Col 2.	Cube of Number in Col. 3.
400	.1366	7.32	393
600	.1164	8.59	632
800	.1078	9.27	848
1000	.1	10.	1000
1200	.0935	10.7	1160
1400	.0892	11.2	1405
1600	.0852	11.74	1617
1800	.0821	12.17	1800
2000	.079	12.66	2030

The third factor covering depth of heater was obtained in a similar manner and the comparative effect of the several rows of coil were found to average as follows:

Depth of heater	Comparative rise in temp.	Reciprocal of Col. 2.	8/N + .24
2 rows of pipe	.247	4.05	4.24
4 "	.47	2.13	2.24
6 "	.67	1.49	1.57
8 "	.82	1.22	1.24
10 "	.96	1.042	1.04
12 "	1.11	.9	.906
16 "	1.35	.88	.74
20 "	1.56	.64	.64
24 "	1.725	.58	.57⅓
32 "	2.	.5	.48

or rise in temp. varies as $\frac{1}{8/N + .24}$

The tests were made with only the velocities and depths of heater above given.

I find on comparison with several other tables giving rise in temperature under stated conditions that the proportionate factors compare as follows; although I do not know all the conditions under which the others were made:

PROPORTIONATE RISE IN TEMPERATURE WITH DIFFERENT HEATER DEPTHS.

Depth of heater	B. S. H.* formula.	I. C. S.† table.	A. B. C. ‡ chart.	Niagara Rad. table.	Prof. Allen table.
1 row	.1213				
2 "	.2358				
4 "	.446	.49			
8 "	.806	.824		.746	.885
12 "	1.104	1.094			1.117
16 "	1.351	1.32	1.358	1.31	1.336
20 "	1.562	1.54	1.54	1.535	1.568
24 "	1.745	1.745	1.745	1.745	1.745
32 "	2.04		2.038		2.02

* B. S. Harrison.
† International Correspondence School.
‡ American Blower Company.

In the above comparison the factors are all proportioned to 1.745 for 24 rows deep.

From the same tables the relative factors for different velocities compare as follows:

Velocity in feet per min.	B. S. H. formula.	I. C. S. table (*a*)	A. B. C. chart.	Niagara Rad. table.	Prof. Allen table.
400	.1357		.132		
600	.1187	.056	.122		
666	.1145			.113	
800	.1078		.108		
833	.1063			.1038	
960		.0654			
1000	.10			.0995	.1086
1166	.095			.0951	
1200	.094	.071	.0956		
1250	.0929				.0933
1333	.091			.0904	
1400	.0894		.0887		
1500	.0874			(*b*) .0874	.0885
1600	.0854		.085		
1700	.0838				(*b*) .0838
1800	.0822	(*b*) .0822	(*b*) .0822		
2000	.0793		.0812		

(*a*) Figured from the International Correspondence School table of heat transmission on basis that temperature rise is proportional to T. U. emission.
(*b*) Unit to which other readings in same column are proportioned.

In using any formula in connection with blast coil performance the assumptions must be made:

1st. That all coil surface, so figured, is prime heating surface (or efficient heating surface).

2nd. That the coil circulates freely —i.e., is neither air nor water bound in any part.

3rd. That the steam supply and condensation return connections are adequate to keep the coils filled with steam and constantly free from condensation and that the construction of the steam and return headers allow sufficient area for same.

In regard to Chart 2 (see October, 1907, issue, THE HEATING AND VEN-

TILATING MAGAZINE) showing B. T. U. given off per square foot of heating surface of coils per hour, per degree difference of temperature, the results were obtained not by weighing the condensation but were computed from the temperature rise of the air, volumes of air handled, etc.

The formula is derived from the formula of rise in temperature. As given in the October article, the factors in the formula were not arranged in their order so as to show the method of derivation.

Properly arranged the formula becomes

$$\underbrace{\frac{1}{\sqrt[3]{V}}\left(T-t\right)^{\frac{1}{8/N+.24}}}_{A.} \times \underbrace{\frac{.35\,V\times 60}{50}}_{B.} \times \underbrace{\frac{1}{1.9\,N}}_{C.} \times \underbrace{\frac{1}{T-t}}_{D.}$$

A—Total rise in temperature of air passing through coils.

B—Total volume of air handled per hour divided by assumed value of 1 B. T. U. Volume measured before entering coils.*

C—Total square feet of heating surface for coil depth, per square foot superficial area of heater group.

D—Difference in temperature between steam in coils and air entering same.

*The value of 50 for a thermal unit was taken at a temperature of about 12° F. for the entering air, that being taken as an average temperature for cold air entering the tempering coil of a blast coil group. If the average temperature of entering air is 12° and the average temperature of air leaving heater case is taken at, say, 120°, then the average temperature at which the thermal unit value should be figured would be 72° F. and the value would be 55.+ instead of 50.

Then the results given on Chart 2 should be reduced 10%.

In fact, to make the readings of Chart 2 exact the rise in temperature must first be figured by Chart 1 and the readings of Chart 2 corrected by the following factors.

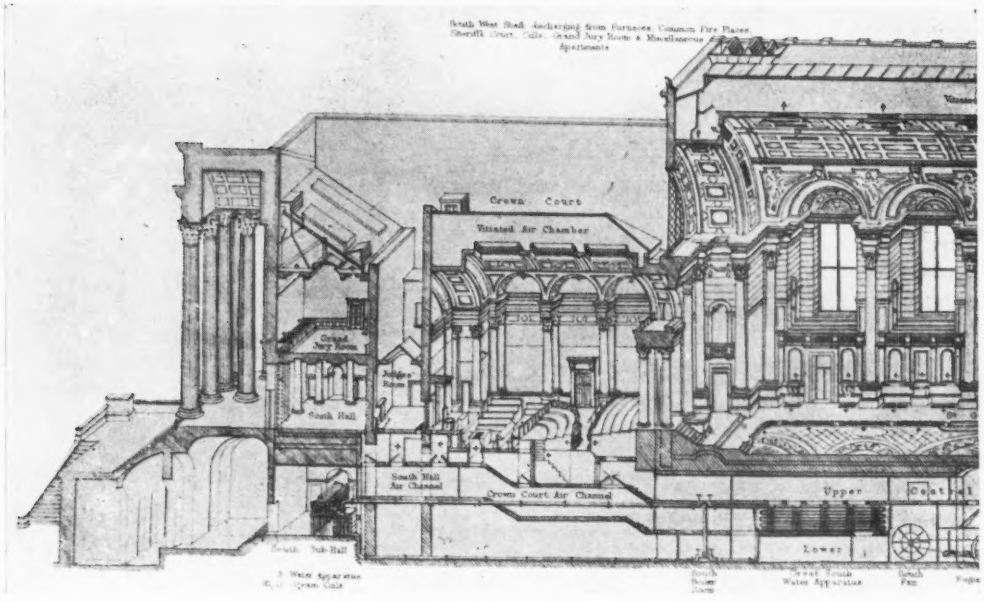
Temp. of entering air plus one-half of total rise in temp. after passing through heater.	Actual value of one B. T. U. in degrees rise of temp. per cu. ft. of air.	Correcting factor by which to multiply reading of B. T. U. given off by coils, from Chart 2.
12° F.	50°	1.
40° F.	52.2°	.9425
60° F.	53.°	.905
80° F.	57.2°	.8725
100° F.	59.4°	.841

BURT S. HARRISON.

The Mechanical Ventilation and Warming of St. George's Hall, Liverpool

II

The accompanying illustration has been sent us by the author, Mr. Charles R. Honiball, of a description of the heating and ventilating system in St. George's Hall, Liverpool, which was published in the October, 1907, issue of THE HEATING AND VENTILATING MAGAZINE. The diagram is taken on a sectional line through the entire building, the principal apartments being shown in isometrical perspective to facilitate the explanation of the ventilating arrangements. The crosses indicate special places in the section where fresh air is admitted and the arrow heads indicate special places where vitiated air is discharged. The diagram will aid



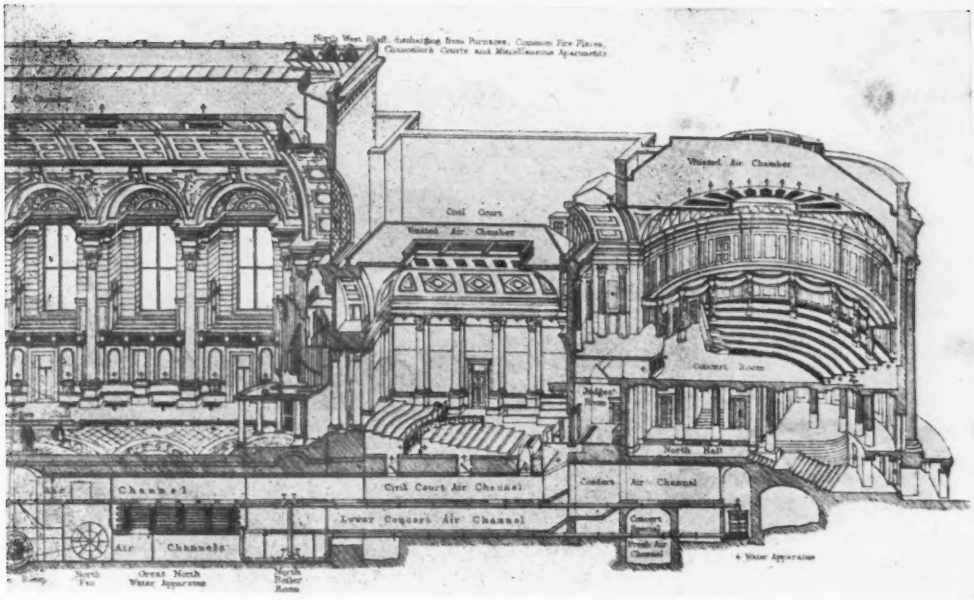
SECTIONAL VIEW, IN ISOMETRICAL PERSPECTIVE, OF ST. GEORGE'S

the reader greatly in studying this interesting system, as described in our October issue.

In his paper, Mr. Honiball said of the system: "There exists to-day no finer example of the genius and skill of that brilliant pioneer of mechanical ventilation, Dr. David Boswell Reid, than the warming and ventilating installation of St. George's Hall, Liverpool. The installation has been in constant work since the opening of the assize courts and St. George's Hall in 1851 and 1854, respectively, and it is working to-day. It has proved entirely successful in working, without occasioning any difficulty at any time. Though it may lack modern labor-saving devices, it possesses many, if not all, the essential features which characterize successful installations of to-day."

Reviewing briefly Mr. Honiball's description of the installation, as published in our columns last month, the apparatus, practically all of which is shown on the accompanying diagram, includes five water coils, named, respectively, the great south hall water apparatus, located in the south sub-hall; the great south water apparatus, the great north water apparatus, the west concert and east concert water apparatus. By their arrangement the whole of the heat can be directed on any of the great apartments, or subdivided, as occasion requires, by opening or clos- the numerous valves and doors in the air chambers that extend all over the building. The boiler rooms, one on each side of the engine room, each contain a hot-water boiler of the Cornish type, 5 feet in diameter by 15 feet long. The flow and return pipes and stop valves to these boilers are 12 inches in diameter. The flow and return pipes and the manifolds for the great north and south water coils are each 10 inches in diameter, while those for coils at the north and south ends of the building are 8 inches in diameter. The hot-water coil pipes are 4 inches in diameter and are made of cast iron.

The steam apparatus, as originally fitted, consisted of 27 coils. One of these will be noticed near the front of the building, in the south sub-hall. They are used principally for heating the chambers, halls and corridors, in which they are placed previous to the commencement of business. They are then shut off when the hot water is found sufficient or sustained. Two steam boilers, each 5 feet in diameter by 15 feet long, supply these steam coils and, in addition, furnish steam



HALL, LIVERPCOL, SHOWING HEATING AND VENTILATING SYSTEM

for working the organ engine, fan engine and an engine for pumping water to a fountain.

The ventilating arrangements were designed to supply from 7 to 10 cubic feet of air per minute to each person in the building. Directly under the middle of the floor of the great hall a 16-horse power engine drives four fans, two of which are shown in the accompanying diagram. These fans are each 10 feet in diameter and 5 feet broad and are driven at a speed of 45 to 60 revolutions a minute.

The space between the floor of the hall and the engine room, it will be noted, is divided into three channels, called respectively, the cold, warm and tempered air channels. Each fan has its own section, but, when necessary, all four fans can be concentrated to supply air to any one section.

In the discharge of vitiated air, owing to the great height of the great hall, the vitiated air from the apartments on each side of it is discharged through the ceiling of the hall into a large vitiated air chamber formed between the ceiling and the roof, and afterward into the open air through glass louvres which extend all around the highest part of the roof.

*The Nature of True Boiler Efficiency**

By Walter T. Ray and Henry Kreisinger

Early in the routine work of the Steam Engineering Division of the U. S. Geological Survey, it was realized that the boiler tests made for the purpose of determining the value of coal for steaming purposes meant little unless the boiler performance was subtracted or divided out, so as to get to the efficiency of the grate and furnace. When attempts were made to separate the performance of the boiler from that of the furnace, it was found that little was known of either, which fact made the separation impossible. It was therefore decided to conduct laboratory experiments on small boilers, to determine the laws which govern heat absorption by boilers.

The purpose of this paper is to

*From a paper read recently before the Western Society of Engineers.

present the more important results of these experiments, and to offer a few formulated laws governing the rate of heat absorption by boilers.

HEAT TRANSMISSION

Heat travels from any hot body only to bodies at lower temperature, therefore, any boiler can absorb only that heat which is above the temperature of the water in it; heat below this temperature will not flow into the boiler water and therefore is not available for absorption. Commercial boilers absorb only part of the heat which is available for them; the percentage of the available heat which is absorbed by the boiler is called the true boiler efficiency. This efficiency depends somewhat on the way the heat is presented to the boiler, but chiefly on the construction of the latter. The true boiler efficiency is then defined as the ratio of the heat absorbed by the boiler to the heat which is available for it, counting only that heat available which is above the temperature of the boiler water.

In any steam generating apparatus the heat is evolved by the burning of fuel in the furnace, and is then transmitted through the space and through the water heating plates into the boiler water. The path of the heat travel and the mode of the transmission is shown in Fig. 1.

In practice, the water heating plate of the boiler is always covered on the outside with a coating of soot, and on the inside with a layer of scale or mud. Just on the outside and entangled in the small recesses of the soot coating is a dense film of gas which adheres to the solid. The density of this gaseous film decreases outwards from the solid layer of soot. It is somewhere within this gaseous film where the dry surface of the water heating plate can reasonably be assumed to exist. There is a similar film of steam and water adhering to the layer of scale on the inside of the boiler, which film can be considered to contain the wet surface of the heating plate.

In the diagram it is shown that heat is communicated to the dry surface of the water heating plate mainly in two ways:

(*a*) By radiation from the hot fuel bed and furnace walls.

(*b*) By convection from the moving gaseous products of combustion. By convection is meant here the process of displacing cold molecules from the adhering film of gas by hotter ones from the moving mass of hot gases.

From the dry surface of the heating plate, the heat is transmitted through the layers of gas, soot, metal, scale, and steam to the wet surface purely by conduction. From the wet surface the heat is carried into the body of the boiler water mostly by the convection of the circulating water.

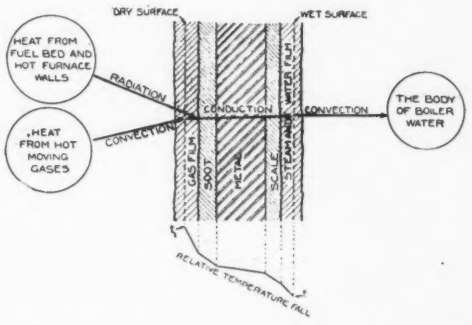


FIG. 1—SHOWING PATH OF HEAT TRAVEL FROM SOURCE OF HEAT TO BOILER WATER

The retardation of any one of these three modes of heat travel lowers the efficiency of the boiler.

It has been said that the dry surface of the water heating plate may be considered as being somewhere in the adhering film of gas. This statement is more correct when it refers to the heat communicated by convection than to the heat imparted by radiation. In the latter case the greater part of the heat passes through the gas film directly to the soot, because gases are to a great extent permeable to the radiant energy.

As the adhering films of gas and the film of steam and water may be, and very likely are, of considerable thickness, the heat must pass through part of the thickness of the film by conduction and as both the gas and steam are very poor conductors of heat, the resistance which these films offer to the passage of heat may be

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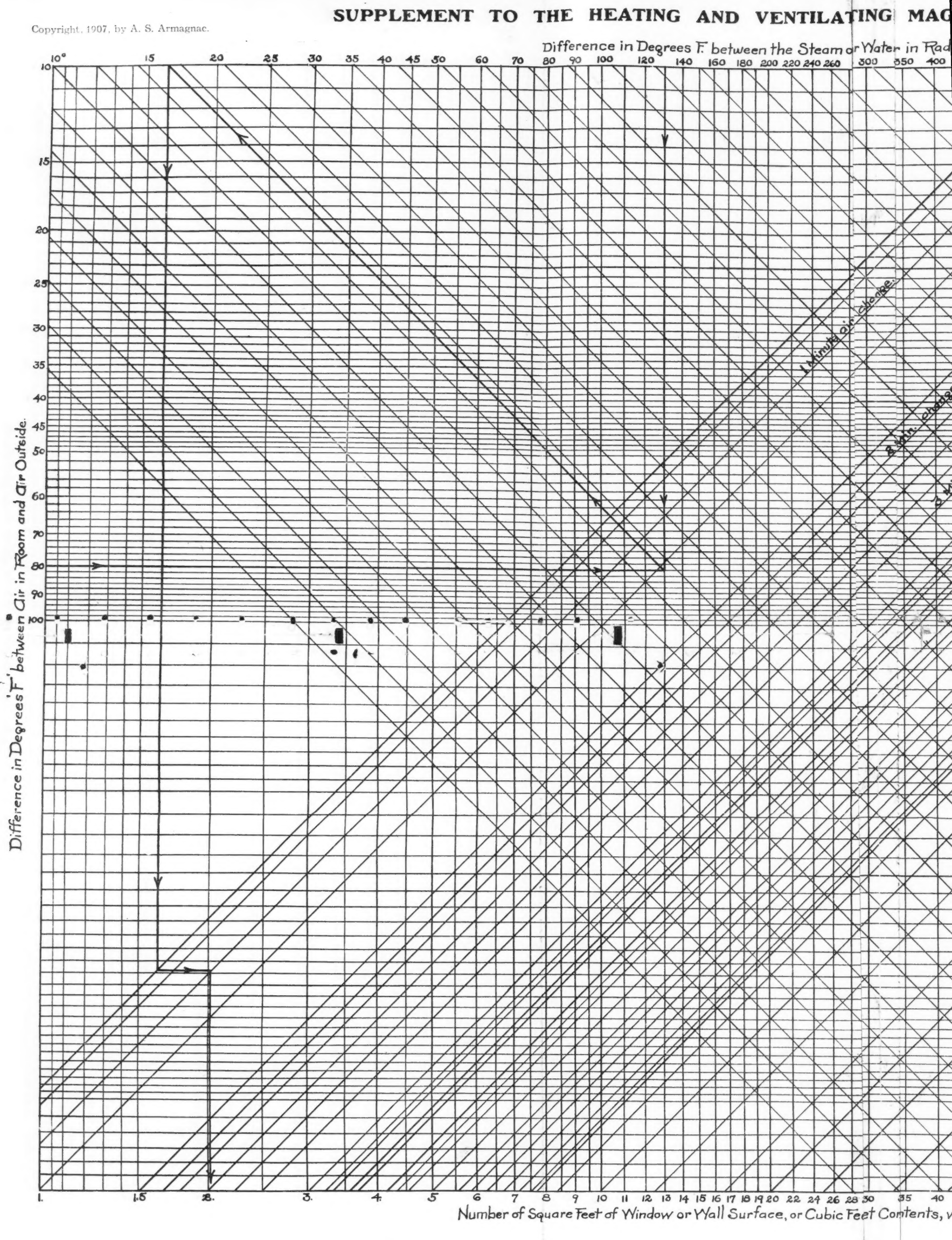
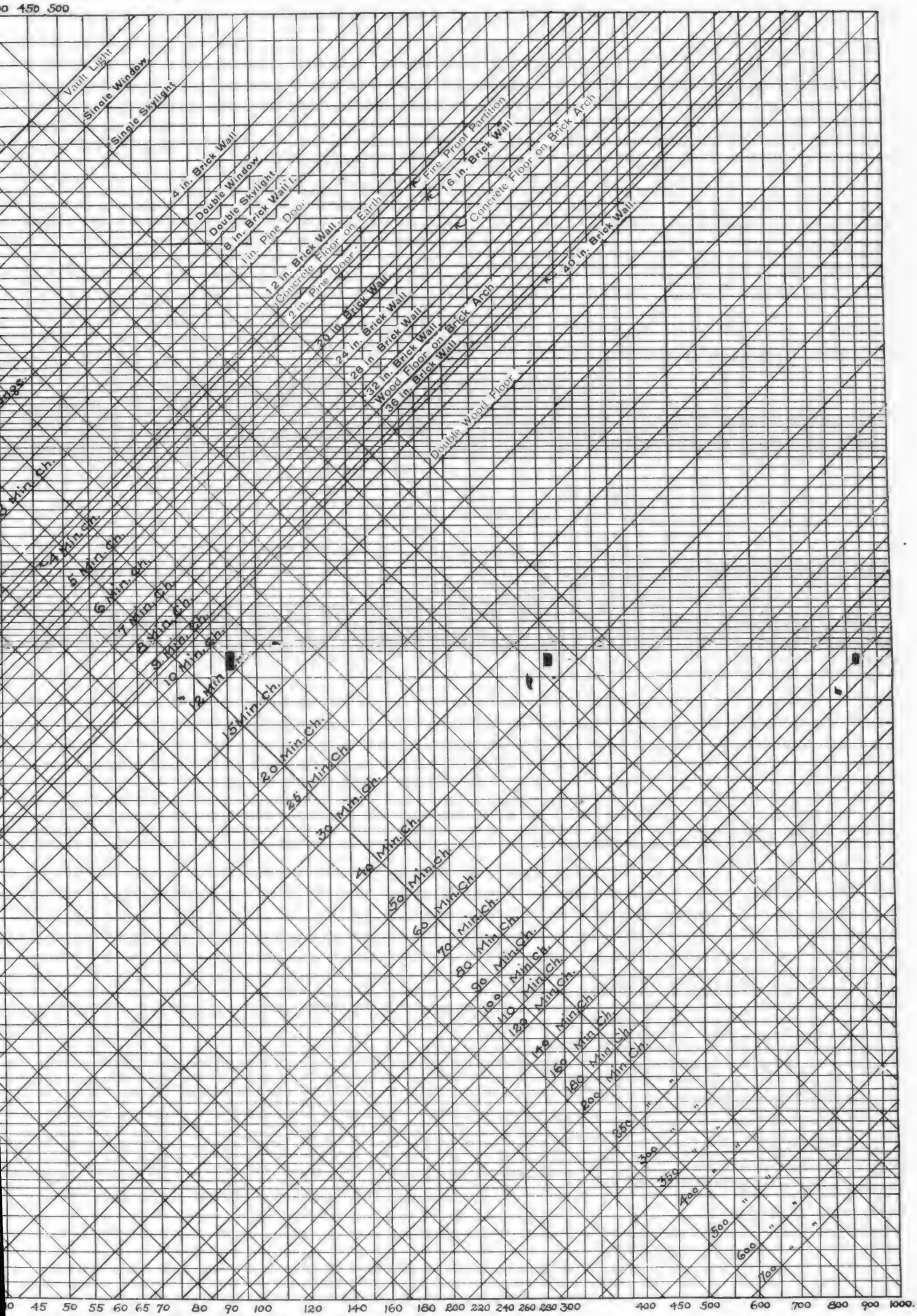

Difference in Degrees F. between the Steam or Water in Rad
10° 15 20 25 30 35 40 45 50 60 70 80 90 100 120 140 160 180 200 220 240 260 300 350 400
10 15 20 25 30 35 40 45 50 60 70 80 90 100
Difference in Degrees 'F' between Air in Room and Air Outside.
1 Minute air change
1. 1.5 2. 3. 4. 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 22 24 26 28 30 35 40
Number of Square Feet of Window or Wall Surface, or Cubic Feet Contents, v

CHART 1.—SHOWING PROPER PROPORTION BETWEEN RADIATING SURFACE

(Accompanying article by Mr. Burt S. Harrison on "Window Leakage and Its Effects on the Amount of Radiation,"

adiator and the Air in Room.



, which One Square Foot of Radiation will provide for.

B. S. Harrison
N.Y.C. 9-16-07.

E REQUIRED AND WINDOWS, WALLS OR CUBIC CONTENTS.

n," appearing in the November, 1907, issue of THE HEATING AND VENTILATING MAGAZINE.)

even greater than the combined resistance of the soot, metal and scale.

Although this paper is intended to discuss mainly the factors which influence the rate of heat impartation by convection, a brief explanation of the laws of the rate of heat radiation and the rate of heat conduction will help in making clear the whole matter of heat absorption by the boiler.

LAWS GOVERNING RATE OF HEAT RADIATION AND OF HEAT CONDUCTION.

The quantity of heat which the boiler receives by radiation from any hot portion of the furnace or the fuel bed may be taken to be proportional to the difference of the fourth powers of the absolute temperatures of the hot parts of the furnace and the soot coating on the boiler plate. This law of radiation is known as Stefan & Boltzmann's Law. Strictly speaking this law applies only to black bodies; however, within the usual temperature range of the boiler furnace it can can applied to boiler problems without any serious error. Mathematically expressed, the net heat transmitted by radiation per unit of area (of hot and cold surfaces) in a unit of time is

$$(1)\quad H = C\ (T' - t')$$

where C is a constant.

T is the absolute temperature of the furnace walls and

t is the absolute temperature of the soot coating on the boiler plate.

It is apparent from this equation that the quantity of heat received by the boiler by radiation increases very rapidly as the temperature of the furnace rises. In boilers where the heat received by radiation is a predominant part of the total heat absorbed, the true boiler efficiency necessarily increases with the rise of the furnace temperature.

The quantity of heat which can be transmitted through a given unit of water heating plate in a unit of time, depends on the difference of the temperatures of the dry and the wet surfaces of the heating plate, and the conductivities of the substance between the two surfaces. This law can be expressed by the following simple formula:

$$(2)\quad H = \frac{c}{d}(t_1 - t)$$

Where H is the quantity of heat transmitted per unit of area of the heating plate, c the average conductivity of the substance between the dry and wet surface, d the distance between the two surfaces, t_1 the temperature of the dry surface, t the temperature of the wet surface.

This law governing the rate of heat transmission by conduction is the same as Ohm's Law used in electrical conduction.

It is evident that with any given condition of the surfaces, the rate of the heat transmission depends entirely on the excess of the temperature of the dry surface over that of the wet surface. For instance, if it is required to transmit double the quantity of heat in the same length of time, the difference of the temperatures of the two surfaces must be doubled. Since the temperature of the wet surface is nearly the same as that of the steam in the boiler and therefore cannot be lowered, the temperature of the dry surface must be raised; as it is this dry surface of the heating plate which cools the furnace gases, the rise of its temperature results in the rise of the temperature of the escaping gases. Thus we see that with the same conditions of the heating plate and the same initial temperature of the furnace gases, the temperature of the escaping gases will rise with increasing capacity. Any cause which, with the same initial temperature of furnace gases, tends to increase the factor $(t_1 - t)$, increases the loss up the stack and lowers the efficiency of the boiler by the same amount.

The main causes of the unnecessarily high factor $(t_1 - t)$ and consequently the cause of high temperature of the waste gases is the presence of the soot and scale on the surface of the heating plate. The heat conductivity of both of these substances is very low, which low conductivity in equation (2) would be indicated by decreasing the value of c, and as the added thickness has

already increased d, it is evident that if H is to remain the same, the factor $(t_1—t)$ must be increased. This last consideration shows that it is important to keep the surfaces of the heating plate free from soot and scale.

The heat imparted to the boiler by convection forms in most cases a large percentage of the total heat received. It is therefore very desirable for the sake of better boiler construction and operation, that the factors which influence the rate of heat impartation by convection be more thoroughly known. Excepting a few experiments done abroad and bearing only indirectly on the steam boiler problem, nothing has been done towards determining these factors. The steam Engineering Department of the U. S. Geological Survey recently stated the investigation of this problem, as an incidental feature in its regular work in testing the quality of coals for steaming purposes. These investigations consist of laboratory experiments made on small models of horizontal multitubular boilers. The laboratory methods and the small boilers were taken up because, *first*, it requires small outlay of money to conduct the experiments, and *second*, it is easier on a small laboratory apparatus to control all the conditions than it would be the case with a large boiler and furnace. It is necessary in work of this kind to keep all the conditions constant.

CONDITIONS OF TEST

Small multitubular boilers were used because they are boilers in the simplest form, the passages of the gas consisting of plain cylindrical tubes. Water tube boilers present much more complicated problems on account of the irregular and variable passages for the gases. The flues were made of thin copper tubes in order to reduce the resistance of the metal to a minimum; as soot and scale were entirely avoided, only the resistance of the film of gas on one side and the film of water and steam on the other was left. An electrical furnace was selected for heating the air, because by its use soot formation and deposition on the heating surface were avoided.

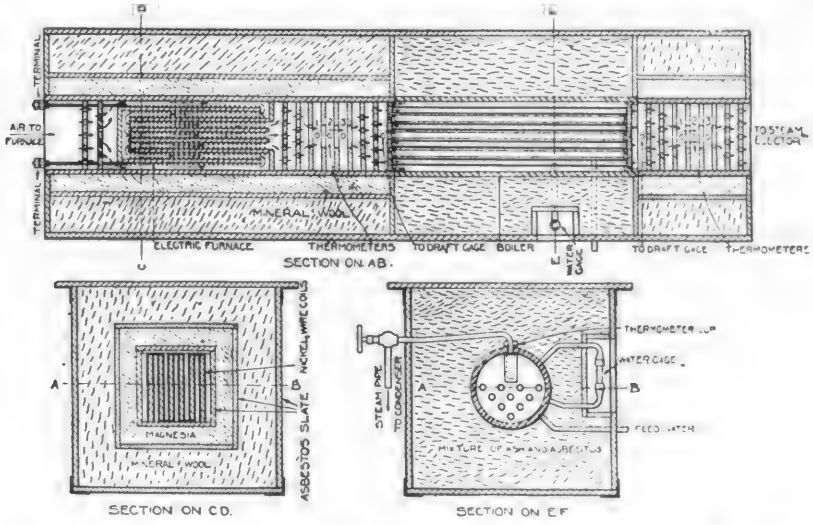


FIG. 2—APPARATUS FOR DETERMINING THE RATE OF HEAT TRANSMISSION THROUGH BOILER FLUES

The apparatus consisted mainly of a small horizontal multitubular boiler, an electric furnace, a surface condensor, and a steam ejector for producing draft. The boiler and the furnace were placed within a box which formed the body of the apparatus. The box was built of asbestos slate and filled with mineral wool and magnesia covering. Fig. 2 shows the assembled view of the apparatus.

So far three boilers were used in the experiments. They were made of a standard 4-inch wrought-iron pipe for the outside shell. The heads were made of ⅛-inch thick sheet copper and were riveted to the 4-inch pipe. The flues were ten in number and consisted of thin drawn copper tubing; they were either soldered or expanded into the boiler heads. Fig. 2 shows two cross sections of one of the boilers.*

evaporated in shorter time to make tests as accurate as when running two hours with low drafts and low temperatures.

INTERPRETATION OF CHARTS

Charts No. 1 to No. 8 were plotted to show graphically the relation between the various factors and the rate of heat absorption. The fact that in some cases the points are somewhat scattered from the smooth curve

DIMENSIONS OF BOILERS

	No. 1	No. 2	No. 3
Length of boiler, outside to outside, of boiler heads	8.28 in.	8.28 in.	16.125 in.
Actual outside diameter of flues	0.252 "	0.313 "	0.252 "
Actual inside diameter of flues	0.175 "	0.230 "	0.175 "
Thickness of the walls of flues	0.033 "	0.041 "	0.033 "
Total area of the inside cross section of flues	0.00167 sq. ft.	0.002885 sq. ft.	0.00167 sq. ft.
Total heating surface of flues	0.31617 "	0.41554 "	0.6156 "

The arrangement of the coils in the furnace is shown in two views in Fig. 2.

The heat absorbed by the boiler was measured by the water evaporated which was condensed and measured. No attempt was ever made to make corrections for moisture in the steam.

The electrical energy used in heating the air was measured by an integrating wattmeter.

Before starting a test, the apparatus was brought to the temperature at which the test was to be run. After a test was started all conditions were kept as nearly uniform as possible. Two to five series of tests were run with each boiler. All tests in each series were of nearly the same initial temperature. Each series contains from 15 to 30 tests which were run with different drafts. Thus each series shows the effect of the velocity of air passing along the heating surface on the rate of heat absorption; the different series of the tests on the same boiler show the effect of the initial temperature; while the tests on different boilers give the influence of the diameter and length of the flues.

The duration of each test was from 1 to 2 hours. With high drafts and high temperatures the tests were shorter, because enough water was

passed through them may be accounted for by errors in temperature measurements. These errors in the worst cases may amount to 5% of the temperature recorded.

The upper curves of Chart No. 1 show that the heat absorbed by the boiler per second varies almost directly as the calculated velocity of air. Tests having nearly the same initial temperature fall in the same smooth curve which is almost a straight line. With the same initial velocity of air, tests with higher temperatures give higher rate of heat absorption. However, the rate of heat absorption is not a straight line function of the initial temperature as indicated by the constant velocity curves of Chart No. 2. As the initial temperature of the air rises, the rate of heat absorption does not increase in proportion to the temperature, and the constant velocity curves flatten down until probably at very high temperature there is very little or no gain in the heat absorbed by further rise in temperature. This fact indicates that there is another factor influencing the rate of heat absorption, which factor varies inversely as the temperature. This factor is the density of the gas. Points for the curves of Chart No. 2

* The authors also included in their paper a detailed description of the rest of the apparatus used in their tests.—EDITOR.

were obtained from the upper curves of Chart No. 1.

The lower curves of Chart No. 1 show that as the initial velocity of air increases, the true boiler efficiency drops at first very rapidly, but after a certain velocity is reached it becomes nearly constant with the low initial temperatures, and varies but little with higher initial temperatures.

The velocity at which for any initial temperature the true boiler efficiency curve becomes nearly horizontal may be called the critical velocity. As shown by the curves, the critical velocity varies with the initial temperature of the air; the higher the initial temperature the higher the critical velocity.

With higher initial temperature the critical velocity is not so well defined as it is with initial temperatures of 500° and 700° F. For very high initial temperatures such as sometimes used in boiler practice (2500° to 3000° F.) the knee probably extends over a considerable range of velocity, so that it is possible that in such a case the boiler never works on the horizontal portion of the curve. It will be shown later that the critical velocity also varies with the diameter of the tubes.

While with low initial temperatures the portion of the true boiler efficiency curve beyond the critical velocity point is almost perfectly horizontal, with high initial temperatures there seems to be a constant drop even beyond this point. This drop in the efficiency can be accounted for by the increased rate of heat absorption. It has been explained before, that with the same conditions of the heating plate the quantity of heat transmitted through the plate in a unit of time depends upon the temperature difference of the dry and wet surface or the factor (t_1-t) in equation (2). Since the only way to increase this factor is to increase t, the temperature of the dry surface which does the cooling of the moving gases, the latter will leave the heating surface

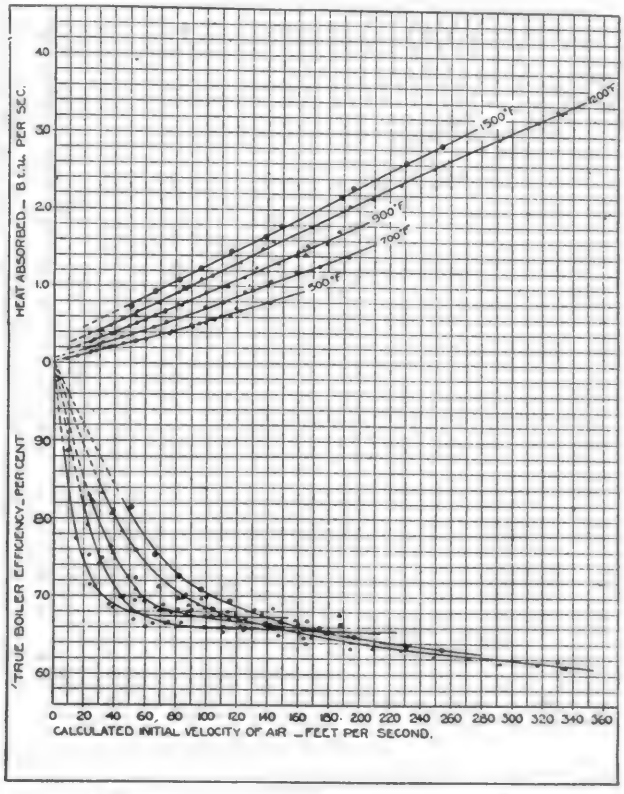


CHART 1—RESULTS OF TEST ON BOILER No. 2

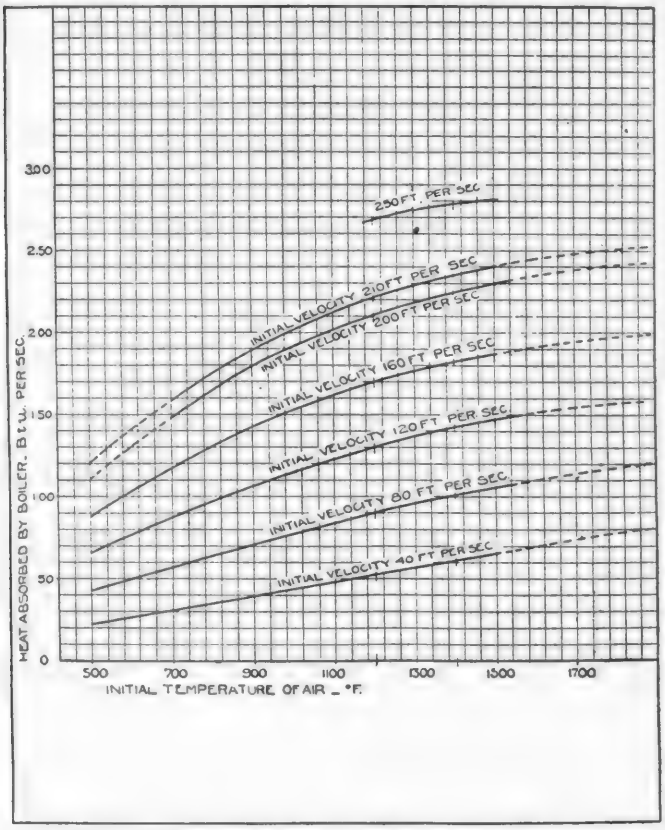


CHART 2—CURVES FROM TEST OF BOILER No. 2 SHOWING THE RELATION BETWEEN THE RATE OF HEAT ABSORPTION AND INITIAL TEMPERATURE OF AIR WHEN VELOCITY REMAINS CONSTANT

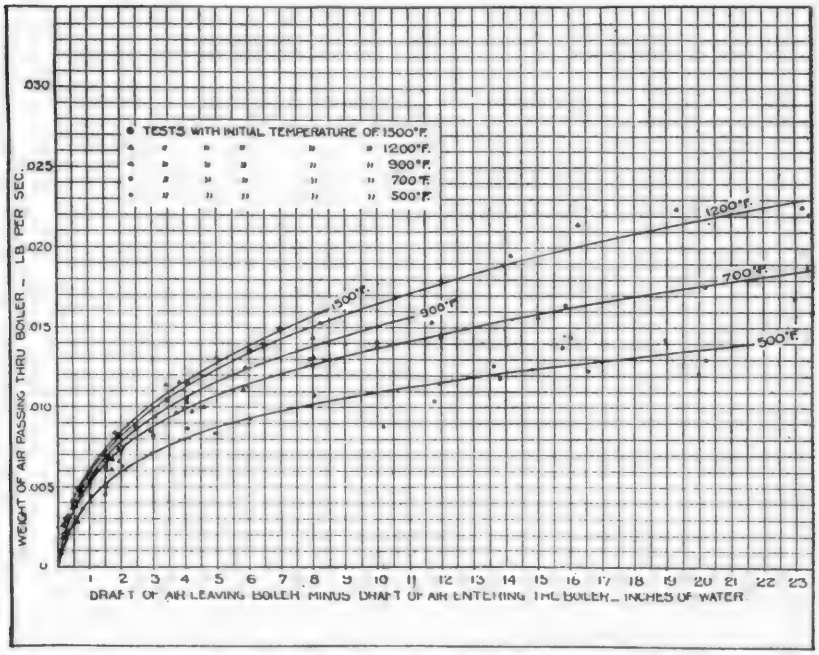

WEIGHT OF AIR PASSING THRU BOILER — LB. PER SEC.
.030
.025
.020
.015
.010
.005
0
TESTS WITH INITIAL TEMPERATURE OF 1500°F.
" " " " " " 1200°F.
" " " " " " 900°F.
" " " " " " 700°F.
" " " " " " 500°F.
1200°F.
700°F.
1500°F.
900°F.
500°F.
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23
DRAFT OF AIR LEAVING BOILER MINUS DRAFT OF AIR ENTERING THE BOILER — INCHES OF WATER

CHART 3 RESULTS OF TEST ON BOILER No. 2, PLOTTED AS A CHECK ON THE CALCULATED VELOCITY USED IN CHART No. 1

of the boiler at a proportionately higher temperature than in the case of lower rate of heat absorption.

Chart No. 3 was plotted as a check on the calculated velocity used in Chart No. 1. With constant initial temperature the weight of air passing through the flues must have some constant relation to the difference of pressures at the two ends of the boiler flues. The curves of Chart No. 3 show that such a relation does exist.

The curves have a shape similar to a parabola of the equation $Y2 = Kx$. With constant initial temperature the true boiler efficiency drops at first very rapidly when the difference of drafts increases, but when the latter reaches about 2 inches of water the efficiency remains nearly constant. The gradual constant drop in the efficiency curves of the higher initial temperatures beyond the 2-inch difference of drafts, may be accounted for by rapidly increasing capacity.

It may be questioned why at the left-hand side the curves were prolonged to a point of 100% true boiler efficiency when the difference of draft or the velocity is zero. This must

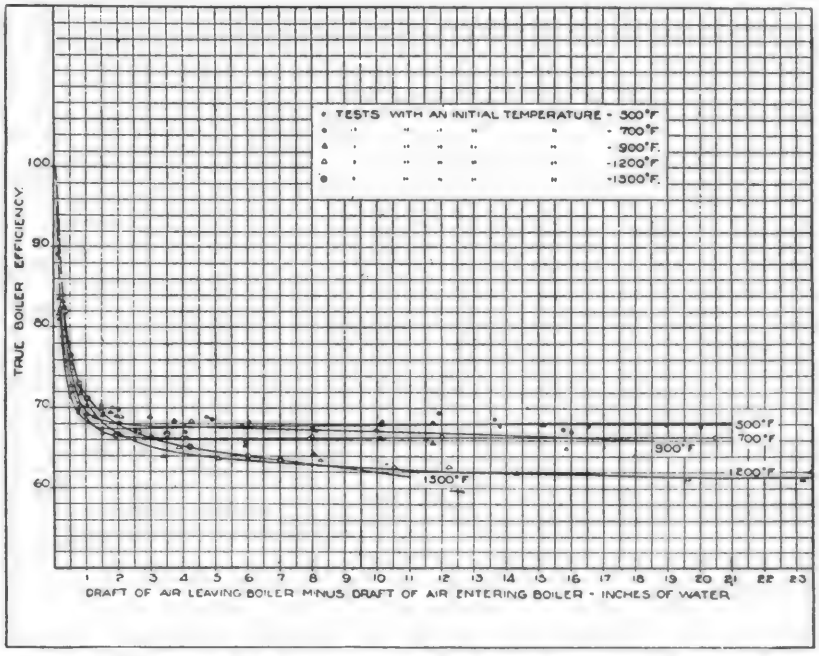

TESTS WITH AN INITIAL TEMPERATURE - 500°F
" " " " " - 700°F
" " " " " - 900°F.
" " " " " -1200°F
" " " " " -1300°F.
TRUE BOILER EFFICIENCY
100
90
80
70
60
500°F.
700°F
900°F
1200°F
1300°F
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23
DRAFT OF AIR LEAVING BOILER MINUS DRAFT OF AIR ENTERING BOILER - INCHES OF WATER

CHART 4—RESULTS OF TESTS ON BOILER No. 4, SHOWING EFFECT OF DRAFT ON TRUE BOILER EFFICIENCY

initial velocity varies directly as the weight of air passing through the flues, and therefore the same relation exists between the initial velocity and the difference of drafts as that shown in the chart to exist between the weight of air and the difference of drafts. The weight of air passing through the flues varies also with the initial temperature, although not in a direct proportion.

Chart No. 4 shows the effect of draft on the true boiler efficiency. The shape of these curves on this chart are somewhat similar to the lower curves of Chart No. 1. The necessarily be so because with no velocity the air would remain in the boiler an infinite length of time and its temperature would finally be reduced to the temperature of the steam in the boiler, making the true boiler efficiency equal to 100%.

As an illustration of rather a small variation in true boiler efficiency for a considerable range in capacity, two sets of tests made on boiler No. 2 are here given. Taking the initial temperature of 1200° F. and the initial velocity of 85 feet per second, the true boiler efficiency is 70% and the rate of heat absorption about 0.96

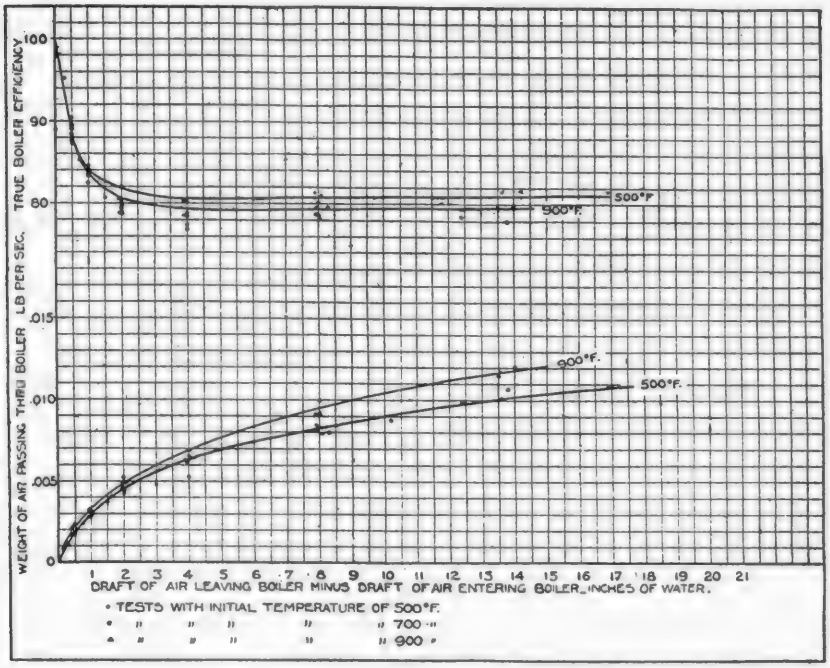

TRUE BOILER EFFICIENCY
WEIGHT OF AIR PASSING THRU BOILER LB PER SEC.
100
90
80
.015
.010
.005
0
500°F
900°F.
900°F.
500°F.
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21
DRAFT OF AIR LEAVING BOILER MINUS DRAFT OF AIR ENTERING BOILER_INCHES OF WATER.
• TESTS WITH INITIAL TEMPERATURE OF 500°F.
" " " " " 700 "
" " " " " 900 "

CHART 5—RESULTS OF TEST ON BOILER No. 1

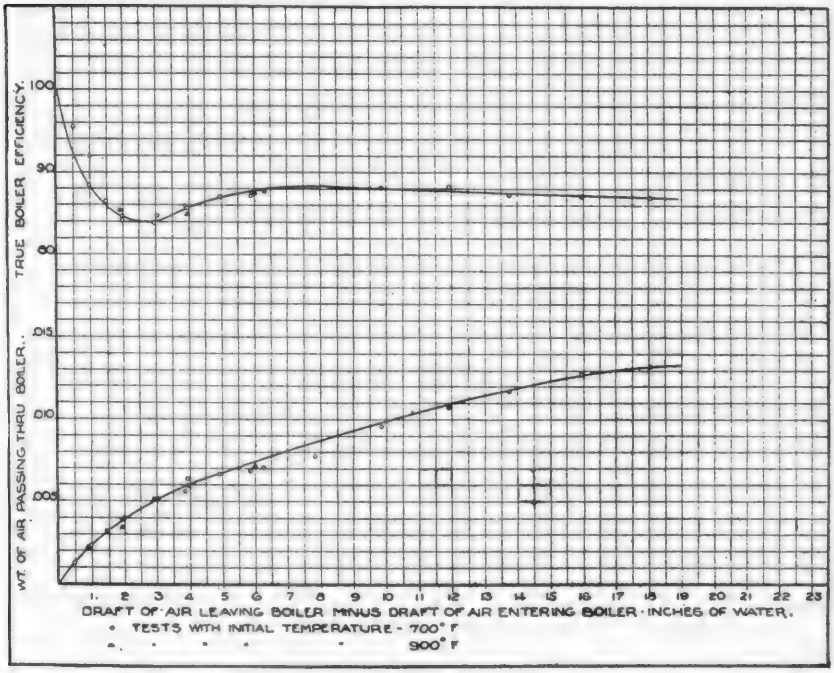

TRUE BOILER EFFICIENCY.
WT. OF AIR PASSING THRU BOILER..
100
90
80
.015
.010
.005
1. 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23
DRAFT OF AIR LEAVING BOILER MINUS DRAFT OF AIR ENTERING BOILER - INCHES OF WATER.
• TESTS WITH INITIAL TEMPERATURE - 700° F
900° F

CHART 6—RESULTS OF TEST ON BOILER No. 3, SHOWING EFFECT OF LENGTH OF FLUES UPON THE TRUE BOILER EFFICIENCY

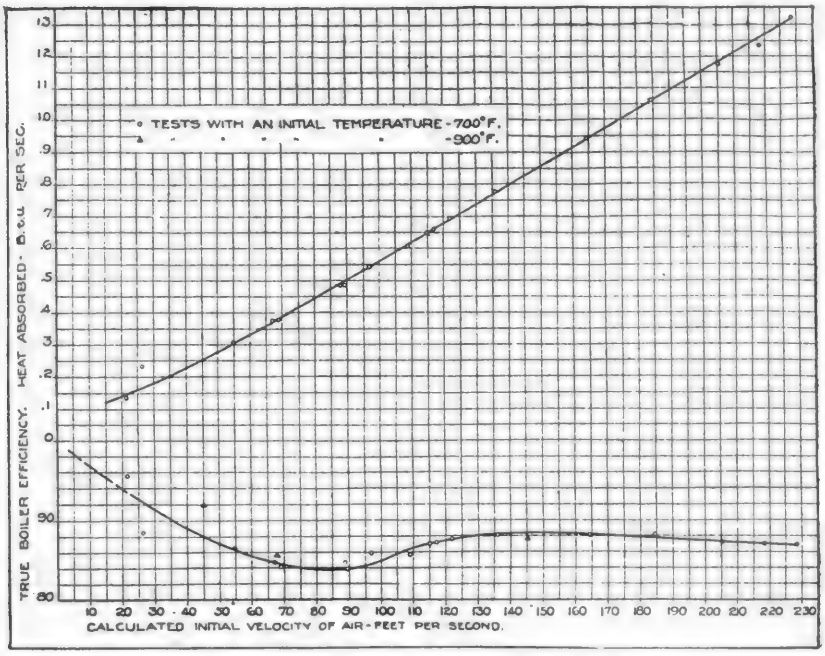

TRUE BOILER EFFICIENCY. HEAT ABSORBED - B.t.u. PER SEC.
TESTS WITH AN INITIAL TEMPERATURE - 700°F.
-900°F.
CALCULATED INITIAL VELOCITY OF AIR - FEET PER SECOND.

CHART 7—RESULTS OF TESTS ON BOILER No. 3

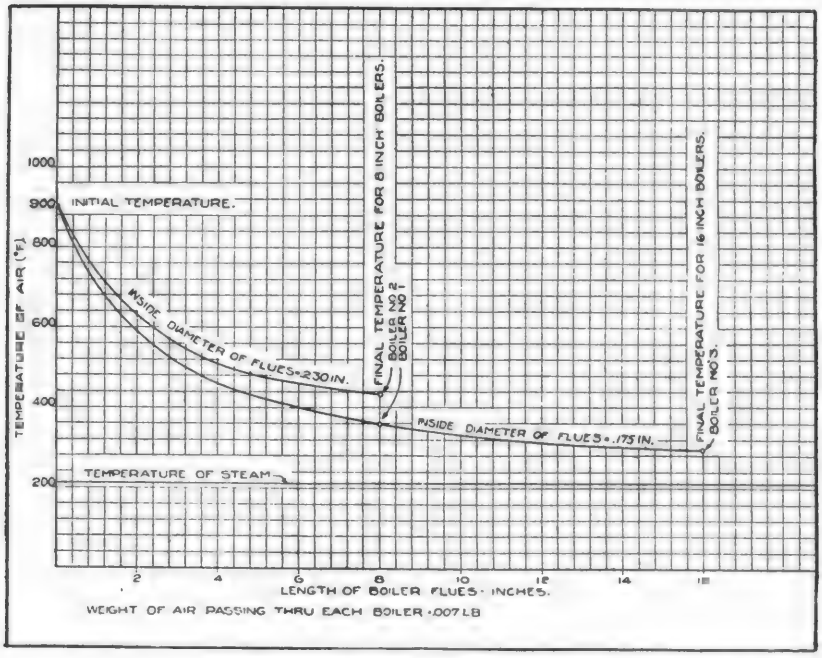

TEMPERATURE OF AIR (°F.)
INITIAL TEMPERATURE.
INSIDE DIAMETER OF FLUES-.230 IN.
FINAL TEMPERATURE FOR 8 INCH BOILERS. BOILER NO 2 - BOILER NO 4
INSIDE DIAMETER OF FLUES-.175 IN.
FINAL TEMPERATURE FOR 16 INCH BOILERS. BOILER NO 3.
TEMPERATURE OF STEAM
LENGTH OF BOILER FLUES - INCHES.
WEIGHT OF AIR PASSING THRU EACH BOILER - .007 LB

CHART 8—SHOWING INFLUENCE OF LENGTH AND DIAMETER OF FLUES ON THE TRUE BOILER EFFICIENCY

B. T. U. per second. Then taking the same initial temperature but an initial velocity of 335 feet per second the true boiler efficiency is about 61.2% and the rate of heat absorption 3.35 B. T. U. per second. In the second case, the capacity has been increased 3.48 times, while the efficiency dropped only 8.8% or 12.6% of itself.

Chart No. 5 shows the same features for tests run on boiler No. 1, as Charts No. 3 and No. 4, show for tests run on boiler No. 2. Comparing the lower curves of Chart No. 5 with the curves of Chart No. 3 it will be noticed that on account of the flues being larger less draft is required for boiler No. 2 than for boiler No. 1 to pull through its flues the same weight of air at the same initial temperature. Thus taking the initial temperature of 900° F., a difference of drafts of 2 inches of water is required to pull through the flues of boiler No. 1 0.005 pounds of air per second, while only $\frac{3}{4}$-inch difference of drafts is required to pull the same amount through the flues of boiler No. 2. On the other hand, small flues have the advantage of being more efficient. As indicated by Charts Nos. 1, 4 and 5, the true boiler efficiency of boiler No. 1 is on the average 14% higher than that of boiler No. 2.

Charts No. 6 and No. 7 give plotted results of tests made on boiler No. 3, which had the same size of flues as boiler No. 1, but its length was double that of No. 1. Tests on this boiler No. 3 were intended to show the effect of the length of the flues upon the true boiler efficiency.

The two Charts, No. 6 and No. 7, show that by doubling the length the efficiency was raised only about 8%.

On comparing the lower curves of Charts No. 5 and No. 6 it will be noticed that a given difference of drafts will pull through both boilers nearly the same amount of air. This would seem to indicate that most of the resistance is at the entrance into the flues and very little of it in the tubes themselves, so that an increase in the length of the flues increases the total resistances very little.

Chart No. 8 shows the influence of length and diameter of flues on the true boiler efficiency. The chart gives the final temperatures of three boilers when the initial temperature is 900° F. and the same weight of air (0.007 pounds) passes through each boiler per second. The curves indicating the drop of the temperature with the length of the flues are asymptotic to the horizontal line of the steam temperature. The chart shows that although the flues of the larger diameter have more heating surface than the flues of the smaller diameter, they are much less efficient. The higher efficiency of the smaller flues is undoubtedly due to the higher velocity of the air passing through them.

By doubling the length of flues of boiler No. 1, only about 8% was gained in the true boiler efficiency as shown by the final temperature of boiler No. 3 it is evident that it does not pay to increase the length of the flues beyond a certain limit.

RESULTS OF TESTS

The deductions drawn from these experiments may be expressed by the following brief statements:

(A) Beyond the points of the critical velocity the rate of heat absorption by convection by the heating surface of boiler flues is nearly a straight line function of the velocity of the gas.

(B) The deviation from the straight line function becomes larger with the rise of the initial temperature.

(C) The critical velocity rises with the initial temperature of the gases, and drops when the diameter of the flue increases.

(D) The rate of heat absorption increases when the initial temperature rises; it also seems to vary directly with the density of the gas.

(E) Increasing the diameter of flues decreases their efficiency as heat absorbs; increasing the length of flues beyond a certain length increases their efficiency very little.

(F) Most of the resistance to the passage of air through the flues is at the entrance into the tubes, the length of the flues increases the resistance but little.

Window Leakage and Its Effect on the Amount of Radiation

By Burt S. Harrison

In figuring radiation for a building, the leakage of air around the windows is an element which often does not receive proper consideration.

The assumption of a natural air change in a room of once per hour, with an added factor for certain directions of exposure, may or may not be sufficient, depending upon many conditions. Only rooms having the same direction of exposure, and the same *relative* dimensions (glass and wall exposure, and cubic contents), will require *relatively* the same proportion of radiating surface per square foot of glass or per cubic foot of contents.

It would seem that the only way to estimate even approximately how much heating surface should be provided to overcome the effect of wind leakage through windows would be to ascertain the direction and velocity of prevailing winds for the particular building in question, compute the velocity of air leakage due to such a wind velocity and from these data, together with the estimated window clearance, obtain the quantity of air in cubic feet which will enter the room per hour by that source.

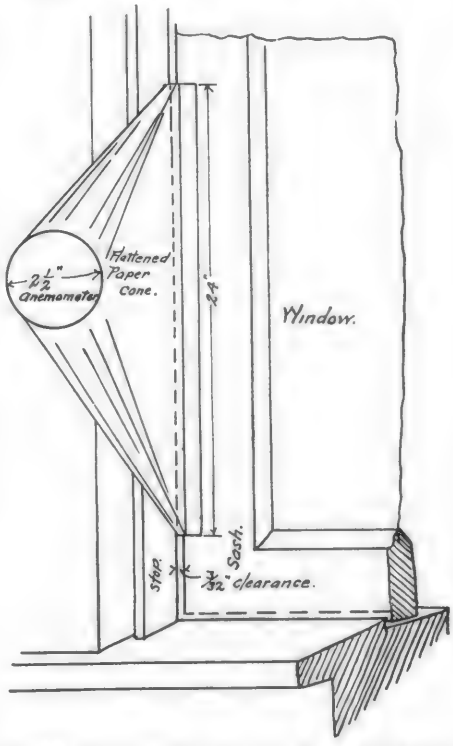
Some years ago, the writer had occasion to make a test of a hot-blast heated school building in which one room could not be heated. The corner exposure which was subjected to a steady wind, and the loose condition of the windows were undoubtedly to blame. In order to determine what the leakage was, the following test was made:

TEST FOR AIR LEAKAGE THROUGH A WINDOW

A pasteboard and paper cone was constructed having the diameter at the small end just sufficient to slip over the wheel case of an anemometer. The other end of the cone was flattened and glued to the window sash and window stop. Wooden wedges were pushed into the crack or clearance space to prevent rattling.

Readings were made from the anemometer and from the average readings the velocity of air entering through the window crack was figured. The sketch showing the arrangement and the results of the tests are given below.

Wind velocity, 15 miles per hour or 1,320 feet per minute. Area of clearance around window, 3-32 inch



APPARATUS DESIGNED BY B. S. HARRISON FOR TESTING AIR LEAKAGE THROUGH WINDOWS, SHOWING METHOD OF OBTAINING ANEMOMETER READING

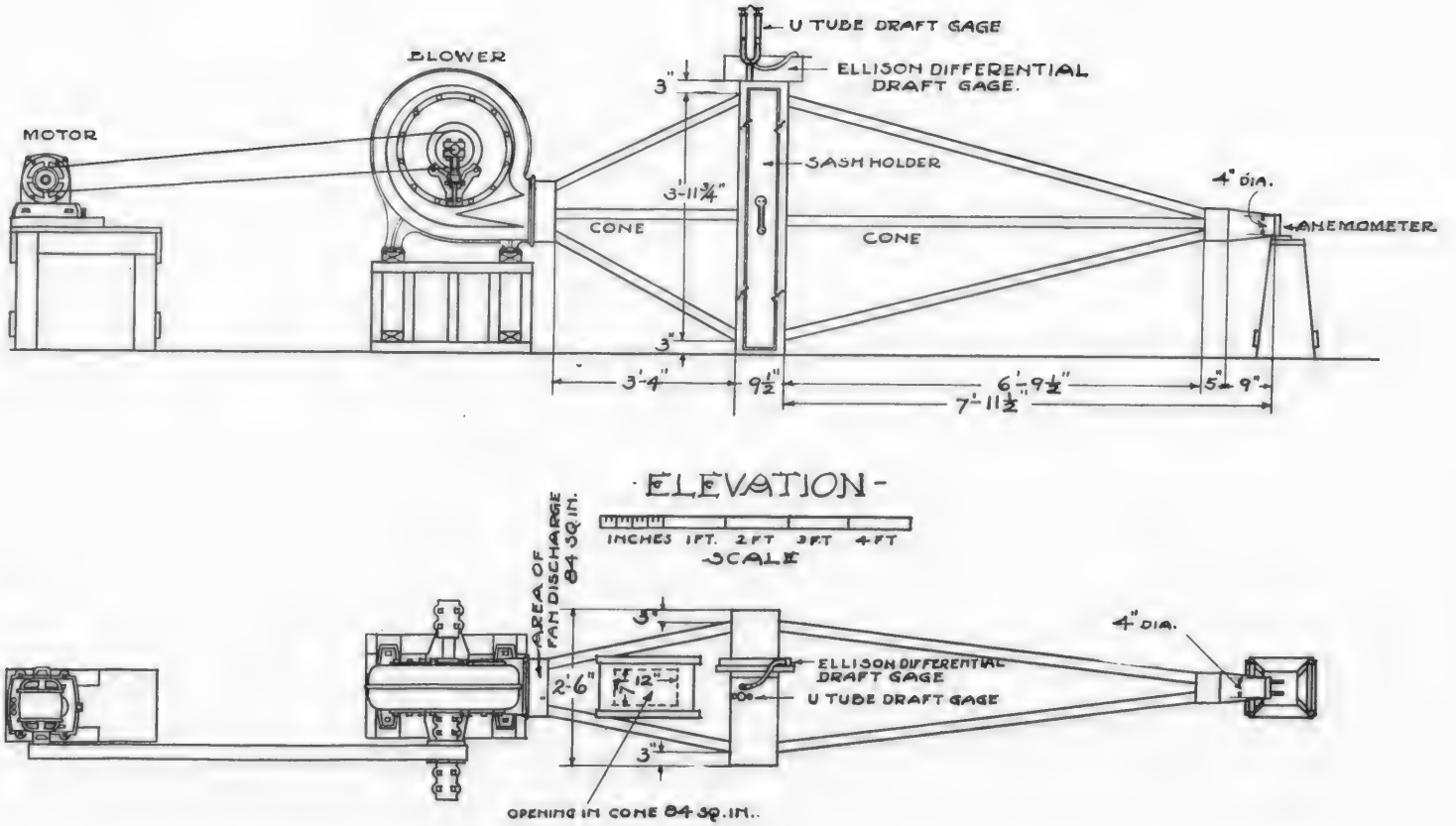
(approximate average). Area of crack included in cone, 3-32 inch x 24 inches = 72 ÷ 32, or 2.25 square inches.

Area of anemometer wheel case 2½ inches diameter = 5^2x3.1416 = 4.9 square inches.

Velocity registered by anemometer (corrected), 85 feet per minute.

Velocity through crack 4.9 ÷ 2.25x 85 = 185 feet per minute.

2.25 ÷ 144x185 = 2.889 cubic feet of air per minute in leakage, or 1.45



APPARATUS DESIGNED BY H. W. WHITTEN FOR TESTING AIR LEAKAGE THROUGH WINDOWS, AIR BEING FORCED AGAINST WINDOW THROUGH ENCLOSED CONE

cubic feet per minute per one foot length of clearance space around the window.

With an ordinary window of the double sash type of average size, 36 inches wide by 72 inches high, the leakage would be approximately:

For 3–32 inch clearance, 30.45 cubic feet air per minute.

For 1-16 inch clearance, 20.30 cubic feet air per minute.

For 1-32 inch clearance, 10.15 cubic feet air per minute.

Assuming that 3 feet x 6 feet is the average size of window units for most buildings and that the windows are tight (not to exceed 1-32 inch clearance), then the leakage per square

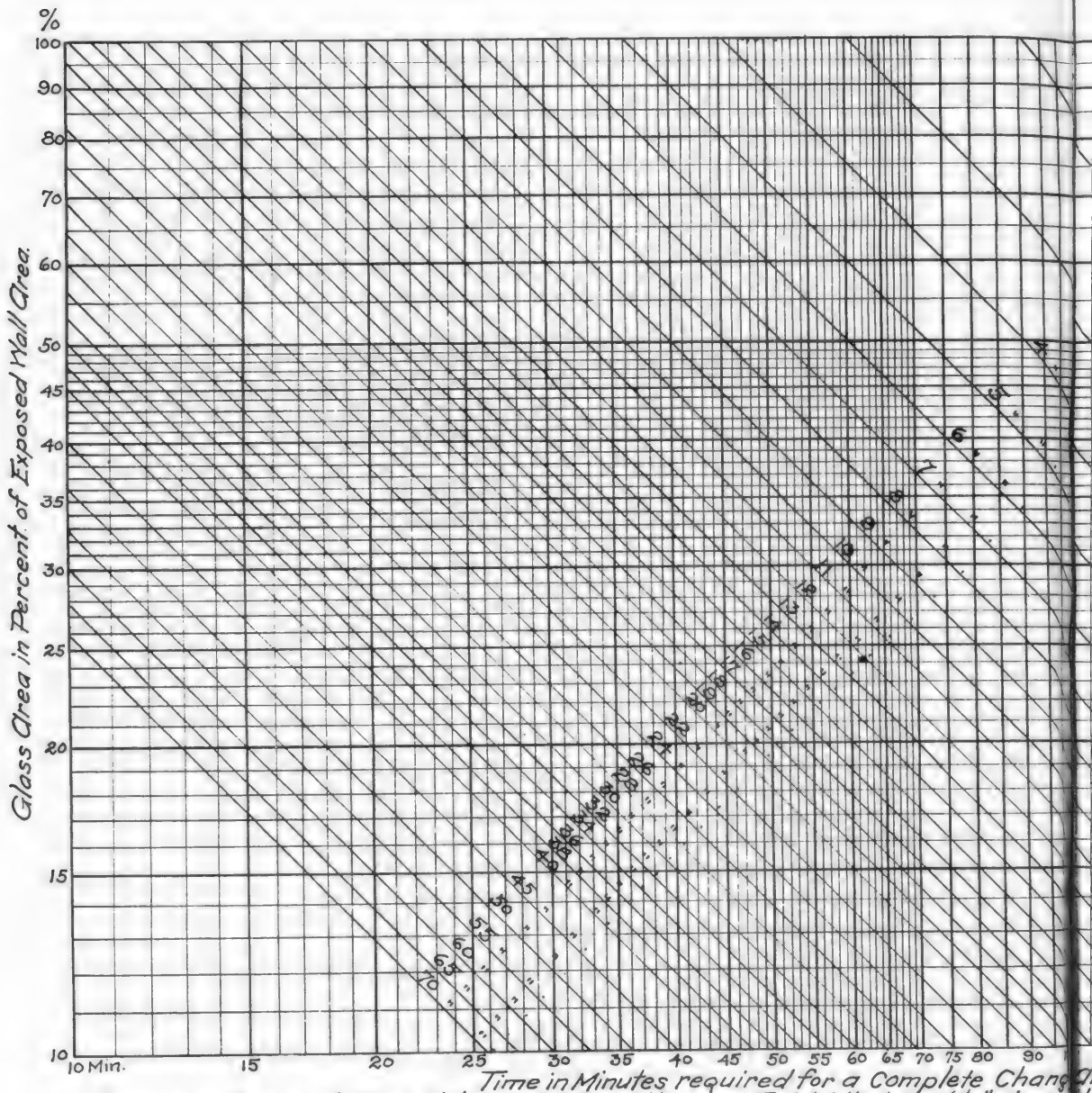


CHART 1.—SHOWING CHANGE OF AIR BY LEAKAGE IN ROOM

foot of glass area with a 15 mile per hour wind is 1015 ÷ 18 or .56 cubic feet of air per minute.

On this basis, Chart No. 1 was plotted to show the comparative time required for air change in rooms under different percentages of outside wall and window exposure, and of different wind velocities.

HOW TO USE CHART 1

Take for example a room 10 feet x 20 feet x 10 feet high, one exposed wall 20 feet x 10 feet high, two windows 3 feet x 6 feet high, and a location subject to an average winter wind of 15 miles per hour.

Cubic contents, 2000 cubic feet.

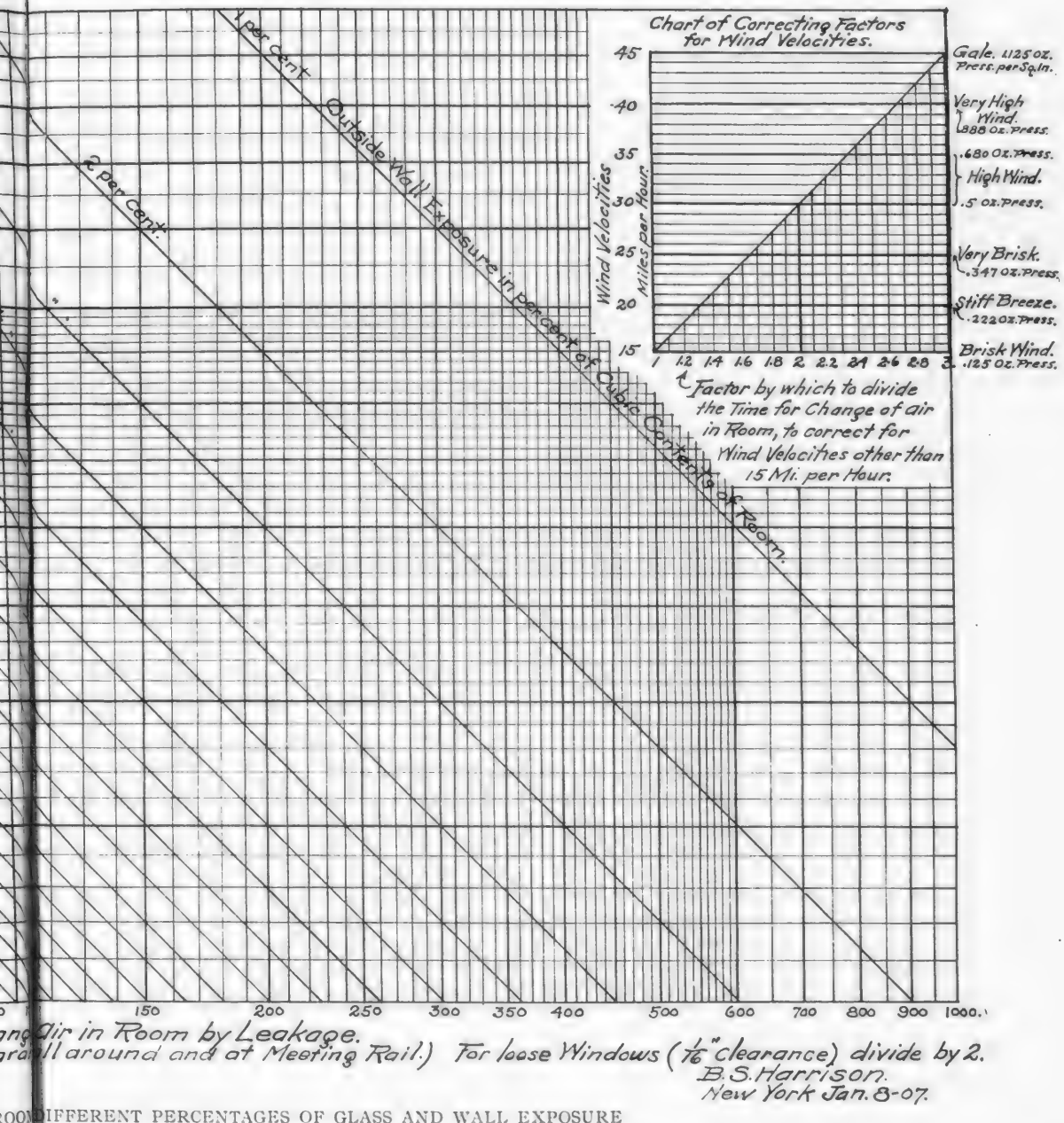


Chart of Correcting Factors for Wind Velocities.
Wind Velocities Miles per Hour.
45
40
35
30
25
20
15
1 1.2 1.4 1.6 1.8 2 2.2 2.4 2.6 2.8 3
Factor by which to divide the Time for Change of air in Room, to correct for Wind Velocities other than 15 Mi. per Hour.
Gale. 1.125 oz. Press. per Sq. In.
Very High Wind. .888 oz. Press.
.680 oz. Press.
High Wind.
.5 oz. Press.
Very Brisk. .347 oz. Press.
Stiff Breeze. .222 oz. Press.
Brisk Wind. .125 oz. Press.
1 per cent. Outside Wall Exposure in percent of Cubic Contents of Room.
2 per cent.
150 200 250 300 350 400 500 600 700 800 900 1000.
Air in Room by Leakage.
all around and at Meeting Rail.) For loose Windows (1/16" clearance) divide by 2.
B. S. Harrison.
New York Jan. 8-07.

ROOM DIFFERENT PERCENTAGES OF GLASS AND WALL EXPOSURE

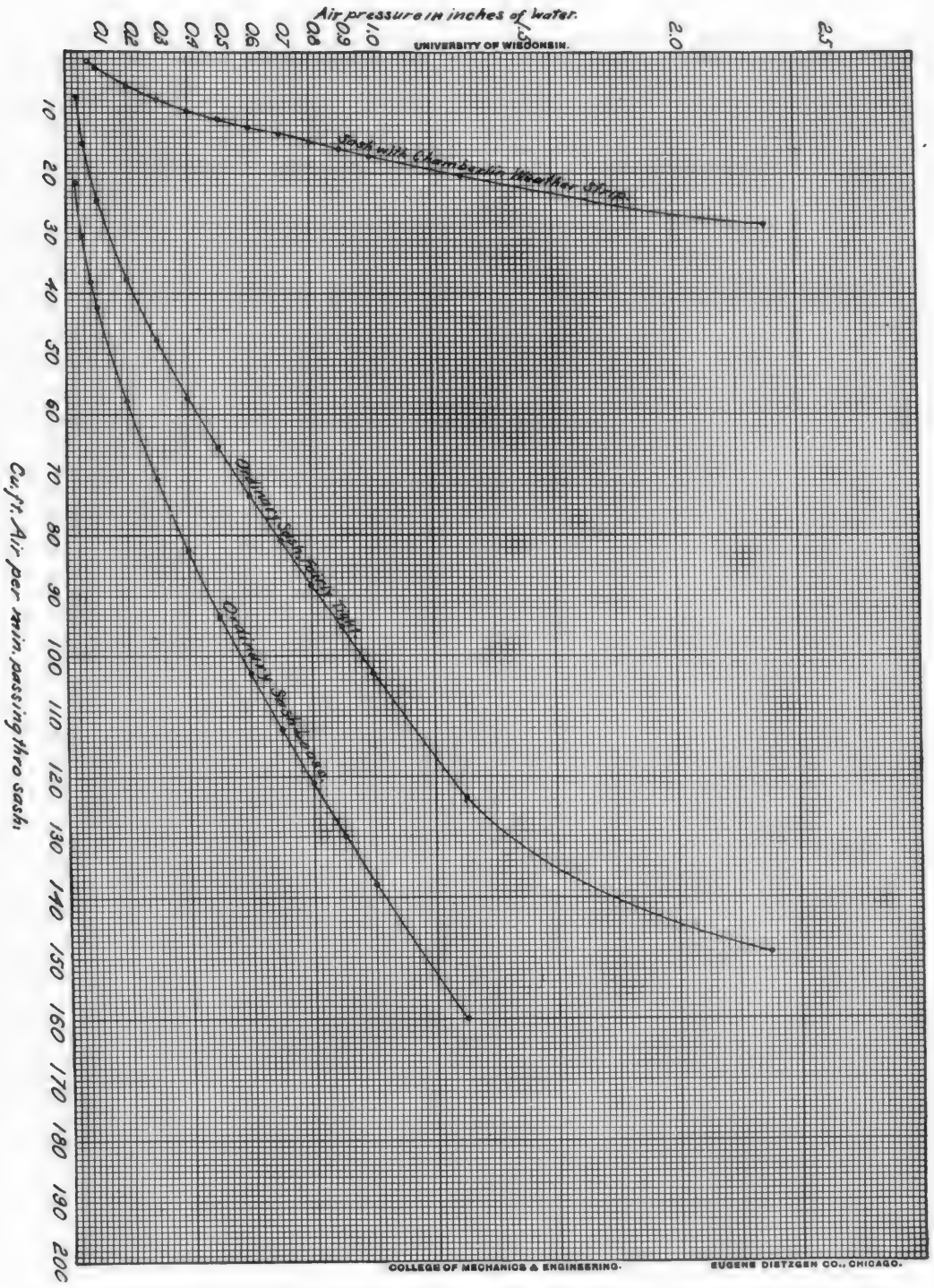

Air pressure in inches of water.
.01 .02 .03 .04 .05 .06 .07 .08 .09 1.0 1.5 2.0 2.5
UNIVERSITY OF WISCONSIN.
Sash with Chamberlin Weather Strip.
Ordinary Sash Poorly Fitted.
Ordinary Sash Well Fitted.
Cu. ft. Air per min. passing thro sash.
10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 170 180 190 200
COLLEGE OF MECHANICS & ENGINEERING.
EUGENE DIETZGEN CO., CHICAGO.

CHART SHOWING AIR LEAKAGES THROUGH WINDOWS UNDER DIFFERENT CONDITIONS, OBTAINED WITH APPARATUS DESIGNED BY H. W. WHITTEN

Wall exposure, 200 = 10% of cubic contents.

Glass, 36 = 18% of wall exposure.

Net wall, 164.

Referring to Chart 1, on the vertical scale at the left of the chart find 18% (glass area in per cent. of wall exposure), follow to the right horizontally until the diagonal line is reached which corresponds to the wall exposure in per cent. of cubic contents (10%). Then follow downward vertically to the base of the chart and the time required for a natural change of air in the room is found to be 100 minutes.

If the window clearance were 1-16 inch instead of 1-32 inch, then the change would take place in 50 minutes. Or, if the wind velocity were 40 miles per hour instead of 15 miles per hour the time for air change would be 100 ÷ 2.662/3 (see chart for correcting factors for wind velocities), or 37½ minutes.

HOW TO USE CHART 2

Having determined the time of air change in the room the amount of radiation required to overcome same can be easily obtained from Chart 2. (See Supplement accompanying this issue.)

*From this chart the proper amount of heating surface for glass and wall exposure and for cubic contents of room is obtainable, whether the source of heat be steam or hot water.

Take the example above given and suppose the plant to be hot water with a temperature in the radiators of 180° F., a desired temperature in the room of 70° F., and an outside temperature of − 10° F.

The difference between water in radiator and air in room = 180—70 = 110° F.

The difference between inside and outside temperature of air becomes 70 + 10 = 80° F.

Find 110 on the horizontal scale at the top of the chart and 80 on the vertical scale at the left-hand side of chart. Follow down vertically from the one point and to the right horizontally from the other, to the point of intersection. Thence follow the diagonal lines, upward to the left to the top of the chart; thence vertically downward to the first diagonal line running upward to the right. From this point proceed horizontally to the right, placing a triangle on this horizontal line.

The intersection of the horizontal line with the diagonal lines representing window, wall, and time taken for change of air in room (as obtained from Chart 1) will indicate vertically downward on the horizontal scale at base of chart how many square feet of glass or wall, or how many cubic feet of contents one square foot of heating surface will take care of.

In this case the constants become as follows:

Single window, 1.75 square feet per square foot heating surface.

8-inch brick wall, 4.7 square feet per square foot heating surface.

Cubic contents, 175 cubic feet per square foot heating surface.

(On basis of 100 minutes air change as shown on Chart 1 for 15 miles wind and window with 1-32 inch clearance.)

The amount of radiation required then would be

Cubic contents, 2000 cubic feet ÷ 175 = 10.4.

Net wall, 164 square feet ÷ 4.7 = 35.

Glass, 36 square feet ÷ 1.75 = 20.6.

Total, 66.0 square feet.

If the window clearance were 3-32 inch instead of 1-32 inch, the air change would be once every 33⅓ minutes and if, in addition, the wind velocity were 30 miles per hour instead of 15 miles, the change would be every 15 minutes. By Chart 2 then, the ratio of heating surface to the cubic contents would be 1 to 26 and the heating surface required would be

Cubic contents, 2000 cubic feet ÷ 26 = 77.

Net wall, 164 square feet ÷ 4.7 = 35.

Glass, 36 square feet = 1.75 = 20.6.

Total, 132.6 square feet.

Obviously, the condition of the windows in a building (whether tight, loose, or loose enough to rattle) will play a very important part, not only in the successful operation of the heat-

*This chart is figured on the basis of 1.5 B. T. U., given off per hr. per sq. ft., of heating surface per deg., difference between steam or water in radiator and air in room.

ing plant, but in the annual cost of its operation as well. Only in extreme cases can the required radiating surface be reduced 30% by remedying the window leakage, but in many cases fully 15% to 20% of the heating surface could be omitted if the windows were made tight.

Recently my attention was called to certain tests made by Mr. Ralph Collamore, engineer with Smith, Hinchman & Grylls, of Detroit, which show that the leakage of a window having 1-32-inch clearance can be reduced about 85% by the application of a suitable weather strip. The arrangement of the apparatus is shown on page 19.

Some of the readings of the test are as follows:

		Ordinary window 1-32-in. Clearance.		Window equipped with Chamberlin metal weather strips.	
Air pressure in inches.	Corresponding wind vel. in mi. per hr.	Vel. of air through sash.	Cu. ft. air p. m. leakage	Vel. of in leakage	In leakage in cu. ft. air p. m
.1	15.11	284′ P.M.	24.7	32′ P.M.	2.8
.4	30.57	660′ P.M.	57.4	110′ P.M.	9.6
.9	45.86	1092′ P.M.	95.	180′ P.M.	15.6
		Leakage with 1-16-in. Clearance.			
.1			42.		
.4			83.		
.9			130.		

The inleakage figures by this test are considerably higher than those used in the chart and it would seem to me that some of the conditions do not conform entirely to those existing in a building. However, the tests were made for comparative effect, with and without weather strips.

The wind velocities were estimated from the pressure registered by the gauge. While these pressures are within the velocity heads at the speeds estimated in Column 2, still there is the lateral pressure due to the enclosing case, which, it seems to me, does not exist to the same degree on the side of a house.

Then, too, if the case on the anemometer side of the window were rectangular with the anemometer set into a circular opening in the center of farthest side, instead of being pyramidal in shape with the anemometer at the apex, I do not believe so high a velocity reading would be obtained at the anemometer.

Unless a building were of peculiar shape and arrangement a slight pressure would be created within its walls which would tend to overcome to some extent the inleakage. However the comparative effect of these weather strips as shown by his tests is well worth considering.

ADVANTAGES IN THE USE OF WEATHER STRIPS

Referring again to the example above worked out by the charts where it was shown that 132.6 square feet of heating surface was required, if these weather strips were applied to the windows, the amount of heating surface could be reduced, according to Mr. Callamore, by about 65 square feet (85% of 77 square feet) leaving the total heating surface 67 square feet instead of 132.6 square feet or a reduction in this case of nearly 50% of the total radiation surface of the room.

In such a case the reduction in first cost would more than pay for the installation of the weather strips, while the annual saving in fuel would be an item of considerable import.

Although the example above taken is unusual, nevertheless, I have found some cases where the conditions were fully as bad.

A case which would more nearly represent the saving affected on an average building by using efficient weather strips would be the following:

Room, 14 feet x 16 feet x 9 feet 6 inch ceiling height.

One side exposed, 16 feet x 9 feet 6 inches.

Two windows, 3 feet x 6 feet.

Window clearance, 1-16 inch.

Wind velocity, 15 miles per hour.

Steam, 212°—room temperature, 70° F.

Outside temperature, 0° F.

Difference between temperature of steam and air in room, 142°.

Difference between temperature of air inside and outside, 70°.

	Proportion of heating surface	Heating surface in square feet
Wall 152 square feet.		
Glass 36 square feet (Single glass window)	1 to 2.5	14.4
Net Wall 116 square feet (8-inch brick wall) ..	1 to 6.$\frac{3}{4}$	17.2
Cubic Contents, 2130 cubic feet (air change once in 52.5 minutes)	1 to 130	16.4
Total		48.0

Wall 7.14% of cubic contents.

Glass 23.7% of wall area.

(By Chart 1—air change = once in 105 minutes for 1-32-inch clearance or once in 52.5 minutes for 1-16-inch clearance. By Chart 2—a 52.5 minute change has a factor of 130.)

If weather strips were used the leakage could be reduced by 85% (according to Mr. Callamore's tests) or 14 square feet of heating surface, leaving the total heating surface 34 square feet or a reduction of a trifle over 29% on the total amount of heating surface.

It would seem to be a fair proposition in specifying a heating system to require that the test of the plant include a test of window clearance, and if a specified maximum clearance (say 1-16 inch) be found to be exceeded, the owner be required to remedy same; or better still specify an efficient weather strip outright and reduce the amount of heating surface of radiators proportionately.

New Boiler Regulations in Massachusetts

Under the rules issued by the new Massachusetts Board of Boiler Inspectors, established this year by the legislature, the maximum pressure allowed on any steam boiler constructed wholly of cast iron must not be over 25 pounds per square inch, and on any steam boiler, the tubes of which are secured to cast-iron headers, not greater than 160 pounds per square inch.

Fusible plugs must be filled with pure tin. The least diameter of fusible metal must not be less than ½ in. except for working pressures of over 175 pounds gauge, or when it is necessary to place a fusible plug in a tube, in which cases the least diameter of fusible metal shall not be less than ⅜ in. Regarding the location of fusible plugs in the different types of boilers, definite directions are given, among which are the following:

In horizontal return tubular boilers, the fusible plugs are to be placed in the back head, not less than 2 inches above the upper row of tubes and projecting through the sheet not less than 1 inch.

In horizontal flue boilers, in the back head, on a line with the highest part of the boiler exposed to the products of combustion and projecting through the sheet not less than 1 inch.

In locomotive type or star water tube boilers, in the highest part of the crown sheet, and projecting through the sheet not less than 1 inch.

In cast-iron sectional heating boilers, in a section over and in direct contact with the products of combustion in the primary combustion chamber.

Directions are also given for the location of fusible plugs in vertical fire-tube boilers, vertical submerged tube boilers, water tube boilers of the Babcock & Wilcox, Sterling, Heine, Robb-Mumford, Almy and Cahall types, vertical boilers of the Climax or Hazelton type, Scotch marine type boilers, dry back Scotch type boilers and economic type boilers.

In addition, rules are given regarding ing strength of rivets and factors of the minimum pressure on boilers, shear-safety for varying ages of boilers.

Movement to Extend Trade

An important movement in an effort to extend the foreign trade of the United States in manufactured products will be inaugurated in Washington, December 5, when Oscar S. Straus, Secretary of Commerce and Labor, will meet the representatives of the chambers of commerce and boards of trade of the leading cities of the country.

Secretary Straus believes, and his belief is shared by many legislators and commercial men, that the United States needs to take prompt and effective measure looking to the marketing of its manufactured products in foreign countries, if it is not to be outstripped by Great Britain, Germany, France and Italy.

THE HEATING AND VENTILATING MAGAZINE

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IN THE calculation of air leakage through windows and the resultant heat losses caused thereby, heating engineers have been accustomed to use certain factors without, apparently, giving a thought to the possibility of materially reducing this leakage. Indeed, it is safe to say that the allowances usually made for exposed glass surfaces are among the least certain or definite, from a scientific standpoint, of all the ratios given for figuring heating work. This is especially true where a plenum system of heating is used, for it is not an uncommon experience, in such cases, to find the air leakage outward through windows far in excess of that expected or considered possible.

Recent tests along this line would seem to confirm the theory that the factors for air leakage through windows may be easily far larger than those in general use. An even more important result of the tests, however, is the really remarkable showing in the lessening of the air leakage when effective weather strips are installed. If anything like the results obtained in the tests can be duplicated in actual practice, it will be within the power of the heating engineer to reduce his window leakages to a degree that will have a very decided bearing on the amount of radiation required.

The tests in question, it should be noted, were made by forcing air upon a window surface through an air-tight cone. This, of course, may be said not to duplicate actual outdoor conditions where the wind, in striking a building, has a chance to sweep by on all sides of the windows.

TO DUPLICATE approximately the conditions of a test made by forcing air upon a window through an enclosed cone, it is apparent that it is necessary to have the wind strike the window at precisely a right angle. Even then, there is the lateral pressure of the wind to be considered, to say nothing of the increased velocity obtained on the other side of the window when the air passes through another cone and through an anemometer located in the small end of that cone. At the same time, it is the pressure of air upon the window that causes the leakage, and if this pressure can be measured, it would seem to be possible to get practical results whether the air were enclosed or free. In the test referred to, draft gauges were placed inside the cone, between the blower and the window. Thus with the pressures known, it is possible to apply them, with proper corrections, to the pressures of various velocities of wind.

THE design of a heat loss chart in the form of that prepared by Mr. Burt S. Harrison, which accompanies this issue of THE HEATING AND VEN-

TILATING MAGAZINE as a supplement, is one that reduces to a handy rule of thumb the well-known formulæ for estimating losses of heat through building materials. Another advantage of its arrangement is that by its use the figures of others in making up estimates of sizes, etc., of material to be used in heating systems can be accurately and quickly checked without the necessity of verifying in detail the estimator's processes of calculation. As a striking illustration of the tendency of the more progressive men in engineering and other professions to share their knowledge with others in the same line of business for the good of all, the chart will, we think, take rank with any recent contribution that has been made to the science of heating and ventilating.

FOR the benefit of those who will want copies of Mr. Harrison's heat loss charts for permanent use in their work, we have printed a limited special edition of the charts, mounted on heavy cardboard, suitable for framing or posting. Each chart is approximately twice the size of the copies published on other pages of this issue. Chart 1 measures 11x21 inches and Chart 2, 21x30 inches. The set will be sent to any address, postpaid, on receipt of $1.00.

IN THE operation of the Interstate Commerce Law, passed by Congress last winter, many shippers claim that the law contains a glaring defect in that the Commission is powerless to prevent imposition of unjust rates until the tariffs are filed, so that the most unfair rate imaginable can be put into effect for a considerable period of time before decision can be had as to its reasonableness. It is claimed that interstate freight rates should not be changed without first giving the shipper the opportunity to be heard. An amendment to the Hepburn law has been proposed, which is being widely circulated and will be presented to Congress during the coming winter. It reads as follows:

"No increase of an interstate rate, or discontinuance of a rate effecting an increase, should be permitted without opportunity to protest being afforded, and upon hearing and determination as to reasonableness when objection is made, in advance of the new rate becoming effective."

THE asphyxiation of a young man in Grand Rapids, Mich., recently by the fumes of a gas heater used in warming water for bathing purposes calls renewed attention to the type of heating apparatus that, as generally installed, provides no means for carrying away the products of combustion. As stated at a recent meeting of the heating engineers, a typical apparatus of this sort, which was used for warming a house, consisted of groups of large gas burners located in the registers in the various rooms. With the rapid growth in recent years in the use of gas and oil for heating purposes, this method of installing them has become widespread and, in many cases, is working great injury to the health of those living in tightly-built houses. It is well known that such products of combustion form a poisonous gas whose effects are deadly and any agitation that can be developed to put a stop to the use of unvented gas or oil heaters would be one in the best interests of the ventilating engineers as well as of the public.

The Weather for October

The accompanying charts and data show the weather conditions during the month of October, 1907, in New York, Chicago, St. Louis, Pittsburg and Boston. It will be noted that in each city from which reports were received heat was required during a considerable portion of the month. In all cases the heating season commenced during the latter part of September.

NEW YORK

In New York, the highest temperature was 73° on the 7th, and the lowest, 36° on the 31st. The greatest daily range was 28° on the 8th and the least daily range, 8° on the 30th. The mean temperature for the month was 52°, 3 below the normal temperature for 37 years. There were 3.82 inches of rain, 1.96 inches falling on the 27 and 28th. The prevailing direction of the wind was northwest and the total movement, 9,227 miles; average hourly velocity, 12.4 miles; maximum velocity for 5 minutes, 50 miles from the northwest on the 29th. There were 18 clear days, 9 days partly cloudy, 4 cloudy and 9 rainy.

BOSTON

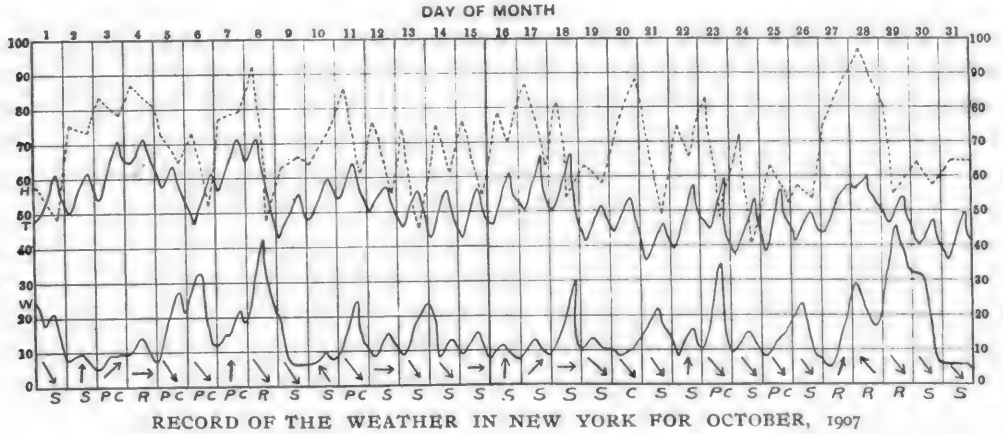
In Boston, the highest temperature recorded was 74° on the 17th, and the lowest, 32° on the 21st. The greatest daily range was 28° on the 18th and the least daily range, 8° on the 30th. The mean temperature for the month was 50° which is 2.5° below the normal temperature for 37 years. The total rainfall for the month was 2.54 inches. The greatest rainfall in any one day was 1.14 inches on the 8th. The prevailing direction of the wind was northwest and the total movement, 7,996 miles; average hourly velocity, 10.7 miles; maximum velocity for 5 minutes, 35 miles from the southwest on the 8th. There were 14 clear days, 5 rainy days, 9 days partly cloudy and 8 cloudy

CHICAGO

The highest temperature recorded for the month was 81° on the 2d and the lowest, 35° on the 28th. The greatest daily range was 29° on the 22d, and the least daily range, 3° on the 30th. The mean temperature for the month was 53°, corresponding to the normal temperature for 37 years. During the month 0.93 inches of rain fell, the greatest amount of rain in any one day being 0.64 inches on the 3d. There were 2.8 inches of snow. The prevailing direction of the wind was southwest; total movement, 10,842 miles; average hourly velocity, 15 miles; maximum velocity for 5 minutes, 41 miles from the west on the 9th. There were 10 clear days, 10 partly cloudy, 11 cloudy and 8 rainy.

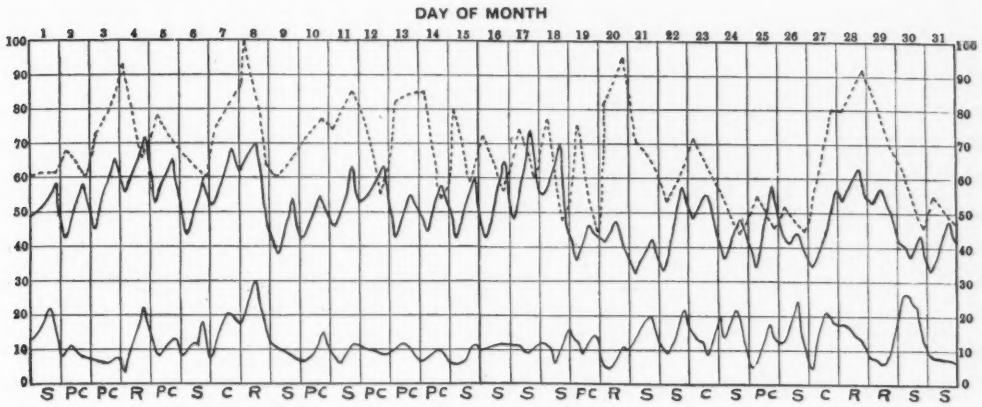
ST. LOUIS

The highest temperature for the

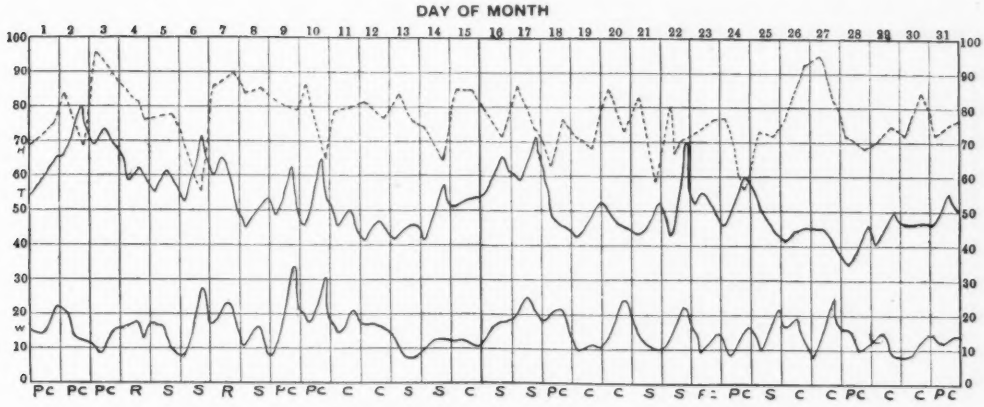


RECORD OF THE WEATHER IN NEW YORK FOR OCTOBER, 1907

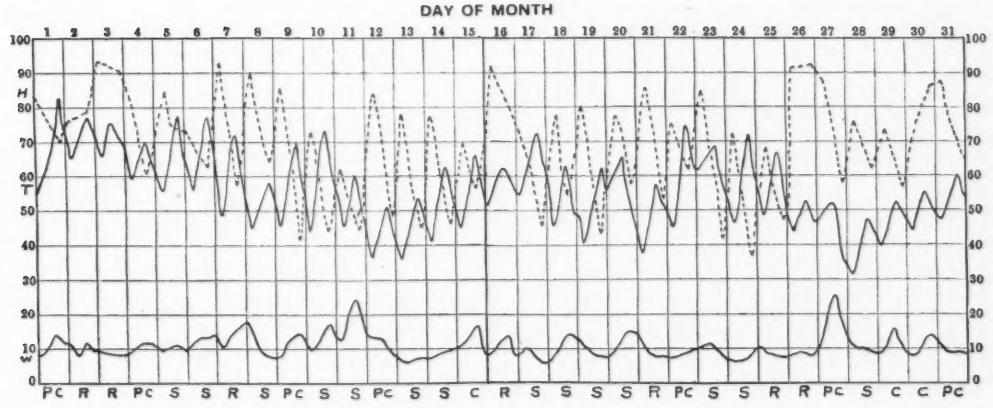
Upper line indicates temperature in degrees Fahr.
Dotted line indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m.
Lower line indicates velocity of wind in miles per hour
Arrows fly with prevailing direction of wind
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain



RECORD OF THE WEATHER IN BOSTON FOR OCTOBER, 1907



RECORD OF THE WEATHER IN CHICAGO FOR OCTOBER, 1907



RECORD OF THE WEATHER IN ST. LOUIS FOR OCTOBER, 1907

Upper line indicates temperature in degrees Fahr.
Dotted line indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m.
Lower line indicates velocity of wind in miles per hour
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain.

month was 82° on the 1st, and the lowest, 31° on the 28th. The greatest daily range was 29° on the 22d, and the least daily range, 9° on the 26th. The mean temperature for the month, 56°, 3° below the average for 35 years. The total rainfall was 3.15 inches, 1.20 inches falling on the 25th and 26th. The prevailing direction of the wind was south; total movement, 7,320 miles; average hourly velocity, 9.8 miles; maximum velocity for 5 minutes, 30 miles per hour from the northwest on the 27th. There were 15 clear days, 8 partly cloudy, 8 cloudy and 8 rainy days.

PITTSBURG

The highest temperature recorded for the month was 79° on the 3d and the lowest, 31° on the 24th. The greatest daily range was 33° on the 2d, and the least daily range, 3° on the 28th. The mean temperature for the month was 50°, 5° below the average for 38 years. The total rainfall was 1.64 inches. The greatest rainfall in any one day occurred on the 3d and 4th, when 0.59 inches were recorded. The average rainfall for 26 years is 2.37 inches. The prevailing direction of the wind was northwest; total movement, 6,902 miles; average hourly velocity, 9.3 miles; maximum velocity for 5 minutes, 42 miles per hour from the west on the 11th. There were 14 clear days, 8 partly cloudy, 9 cloudy and 9 rainy.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

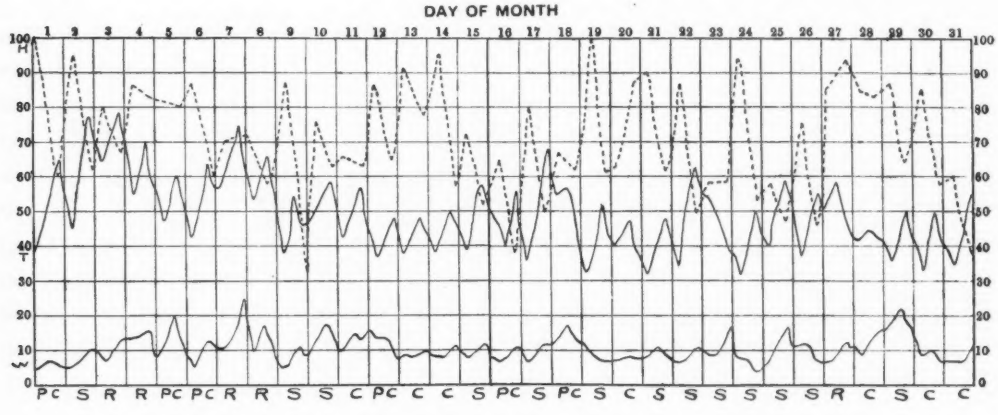
Information Wanted by Committee on Tests

In a communication addressed to the members of the Committee on Tests, through its chairman, Prof. James D. Hoffman, the committee asks for information relating to test data to be included in its report at the annual meeting of the society in January next. All data referring in any way to tests made upon any part of any heating or ventilating system, which data can be used to standardize more completely the method of installation of the apparatus or the testing of it after it is installed, will be acceptable to the committee. The members of the Committee on Tests, to any one of whom the desired information may be sent, are: James D. Hoffman, chairman; William G. Snow, S. G. Neiler, C. R. Bradbury and R. A. Widdicombe.

Annual Meeting of Illinois Chapter

At the annual meeting of the Illinois Chapter of the Society, held October 14 in the rooms of the Western Society of Engineers, Monadnock Block, Chicago, the following officers were elected for the ensuing year: President, James Mackay; vice-president, George Mehring; secretary, John F. Hale; treasurer, August Kehm; board of governors, E. F. Capron, S. R. Lewis and T. J. Waters. A committee on tests was appointed, consisting of R. A. Widdicombe, chairman, Charles F. Newport and M. L. Patterson.

Meetings of the Illinois chapter have been arranged for the second Monday of each month from October to May. The November meeting was held November



RECORD OF THE WEATHER IN PITTSBURG FOR OCTOBER, 1907

Upper line indicates temperature in degrees Fahr.
Dotted line indicates relative humidity in percentage from readings taken at 8 a. m. and 8 p. m.
Lower line indicates velocity of wind in miles per hour.
S—Clear. PC—Partly Cloudy. C—Cloudy. R—Rain.

11 when Mr. L. Brush delivered a stereopticon illustrated talk on "The Manufacture of Radiators and Boilers."

American Society of Mechanical Engineers

The society's fifty-fourth annual meeting will be held in the Engineering Societies Building at 29 West 39th Street, New York, December 3-6, 1907. Symposiums on foundry practice, giving the experiences of prominent men in that work, have been arranged. The specific heat of superheated steam will also be taken up. The utilization of low-grade fuels in gas producers, combustion control in gas engines, tests of producer gas engines, etc., will be given a session. Other live topics, such as industrial education, power transmission by friction driving, cylinder port velocities, etc., will be discussed.

Announcement is made that hereafter the reference libraries of the American Institute of Electrical Engineers, American Society of Mechanical Engineers and American Institute of Mining Engineers, in the Engineering Society's Building, at 29 West 39th Street, New York, will be open evenings until 9 P. M. on all week days except public holidays. These libraries, constituting practically one library of engineering, are available to members of the three societies, and to other engineers and the public generally, under certain rules. Strangers are requested to bring letters of introduction from members or to secure cards of introduction from members, or to secure cards from the secretaries of the respective societies.

American Foundrymen's Association

Toronto, Canada, has been designated as the place for holding the association's next convention, the first week in June, 1908.

British Institution of Heating and Ventilating Engineers

At the Autumn meeting of the Institution held October 8 last in London the following officers were elected for the ensuing year: President, A. B. Simpson, who has served as senior vice-president of the Institution; C. I. Haden, W. Yates and J. D. Sutcliffe (all re-elected). C. J. Seward and J. G. Walker were nominated as vice-presidents, to be balloted for subsequently. The mid-winter meeting will be held February 18 in London, the summer meeting June 23 and 24 at a point not yet selected and the autumn meeting October 6.

Mr. Simpson, the new president of the Institution, is a director of the well-known heating firm of James Simpson & Co. (Ltd.) of London and Newark-on-Trent.

Eastern Supply Association

At the annual meeting of the Eastern Supply Association, held October 9, at the Hotel Astor, New York, the following officers were elected for the ensuing year: President, Henry Aird, Aird-Don Co., Troy, N. Y.; first vice-president, John A. Murray, New York; second vice-president, Frank H. Locke, Locke, Stevens & Co., Boston; treasurer, F. A. Bucknam, Nason Manufacturing Co., New York; secretary, Frank S. Hanley, 261 Broadway, New York. Board of directors: Henry Aird, John A. Murray, F. H. Locke, F. H. Bucknam, J. P. Fell, W. A. Case & Son Mfg. Co., Buffalo, N. Y.; Walter Walls, Walls, Owen & Stambach Co., Philadelphia; John McPhail, James Robertson Mfg. Co., Baltimore, Md.; W. L. Blake, W. L. Blake & Co., Portland, Me.; John McClure Chase, R. C. McClure Co., Syracuse, N. Y., and McCambridge & Co., Philadelphia; G. R. Adams, Samuel Sloan & Co., Rochester, N. Y.; and John Reid, J. L. Mott Iron Works, New York.

Central Supply Association

New officers were elected as follows at the annual meeting of the Central Supply Association, held in Chicago, October 23: President, S. H. Moon, Ahrens & Ott Mfg. Co., Louisville, Ky.; first vice-president, George H. Gorton, W. B. Young Supply Co., Kansas City, Mo.; treasurer, Edward F. Niedecken, Hoffman & Billings Mfg. Co., Milwaukee, Wis.; secretary, Paul Blatchford, Tribune Building, Chicago. Executive committee: The president, vice-presidents, treasurer and the following: James M. Young, Fort Pitt Supply Co., Pittsburgh; J. B. Rahm, United States Supply Co., Omaha, Neb.; P. J. Frey, Central Supply Co., Minneapolis; John D. Hibbard, John Davis Co., Chicago.

An important action taken by the association was its withdrawal, by a unanimous vote, from the so-called Detroit resolutions. The association's adherence to these resolutions had been declared by the Attorney-General of Ohio to be a violation of the Valentine law, in that the association belonged to a trust.

National Committee of the Confederated Supply Associations

Following the meeting of the Central Supply Association, the annual meeting of the National Committee of the Confederated Supply Associations was held in Chicago, October 24, when the committee organized as follows: President, J. J. Ryan, Chicago; vice-president, W. A. Myler, Pittsburgh; treasurer, L. O. Koven, New York; secretary, Frank S. Hanley, New York.

The National Committee ratified the action of the Central Supply Association

in its withdrawal from the Detroit resolutions.

Improved Ideal Sectional Boilers

We present herewith engravings illustrating the new improved Ideal 36-inch steam (open view), and 22-inch water boilers manufactured by the American Radiator Company, which represent the general construction of all their improved boilers of this type. It will be observed by reference to the interior view that in this new method of presenting the water to the fire, the circulation is conducted over the fire in thin bridge-like sheets, providing a constant and rapid passage of a large lateral volume of water which widens within the arch as it ascends. The theory of the manufacturers is that a small body of water will naturally absorb heat more rapidly than a large body; and when the volume is presented in several bridge-like sheets directly over the fire, as in this construction, with ample inlet and outlet to each overhanging water chamber, the result must be a rapid generation of steam in the steam boiler, or a rapid and thoroughly heated circulation in the hot water boiler.

The general construction of this new line of Ideal sectional boilers recognizes the idea which has been thoroughly demonstrated in laboratory tests, by the boilers in operation and under comparison with other constructions,—that nearly the whole success of the boiler depends on the gas-burning chamber. In other words, there must be ample grate area, liberal heating surfaces in which the horizontal and vertical principles are correctly blended, and large gas burning areas immediately at the side of and under these fire-to-water surfaces. Overhanging surface and heating surface generally are very important, but, as coal is the basis for results, and its consumption determines the degree of success, it is manifestly necessary that the gases liberated must be rapidly burned in a large area in which the resultant flames can come in contact with surfaces providing a rapid, upward circulation of the water. This liberal gas-burning area is a prime feature in the improved type of Ideal sectional boilers. The application of this idea is the same in connection with a steam or water boiler, i. e. so far as the prime source of results is concerned.

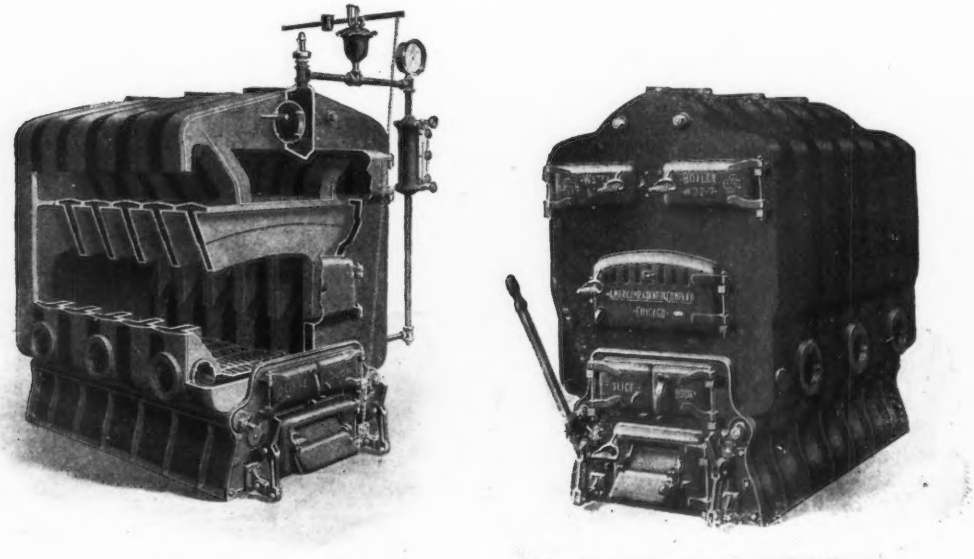
This construction, the merits of which have been proven by practice, tests and practical operation, now characterizes the entire line of Ideal improved sectional boilers with 15, 22, 28, 36 and 48-inch grates.

New Catalogue of Engineering Books

Spon & Chamberlain, 123 Liberty Street, New York, publishers of technical books, have issued their new 200-page catalogue, which will be sent free to those asking for it.

Temperature Regulation for the United States Naval Academy Buildings

The heating and ventilating equipment for the academic group of the United States Naval Academy buildings at Annapolis, described in the October issue of THE HEATING AND VENTILATING MAGAZINE, includes a system of temperature control, applied to all the tempering coils, radiator coils, direct radiators and indirect heating stacks. This system was furnished by the Standard Temperature Regulation Co., of New York.



NEW TYPES OF IDEAL IMPROVED SECTIONAL BOILERS FOR STEAM AND HOT WATER